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IMPROVED CONCEPT OF ELASTOMERIC BEARINGS FOR UH-1 TAIL ROTOR ASSEMBLY

By

C. H. Fagan

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U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

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BELL HELICOPTER COMPANY
A DIVISION OF BELL AEROSPACE CORPORATION
FORT WORTH, TEXAS

U. S. ARMY MATERIEL COMMAND

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DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
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This report presents the results of a follow-on program to Contract DA 44-177-AMC-313(T), "Evaluation of Laminated Elastomeric Bearings in the UH-1 Helicopter Tail Rotor". It includes the design, fabrication, and flight testing of two all-elastomeric bearing configurations for the tail rotor system. The elastomeric bearing configuration offers the advantages of reduced maintenance as well as reduced flapping and oscillatory loads, and provides the designer with a tool to adjust the natural rotor frequencies.

This Command concurs in the conclusions contained herein.

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IMPROVED CONCEPT OF ELASTOMERIC BEARINGS FOR UH-1 TAIL ROTOR ASSEMBLY

Bell Helicopter Report 572-099-008

By

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SUMMARY

This report presents the results of a research program to extend the state of the art of elastomeric bearings and to investigate further their application on a UH-1 tail rotor. In a previous program, elastomeric bearings were shown to be feasible for use on the UH-1 tail rotor flapping hinge (radial bearing) and pitch change axis (thrust bearing). In that program, Teflon journal bearings were used in conjunction with an elastomeric thrust bearing for carrying the edgewise and flapwise bending moments and for stiffness control. For the subject investigation, the Teflon journal bearings were eliminated.

One radial and two thrust bearings were designed and fabricated for the blade grip application. One thrust bearing employed chevron shaped separators, and the other used 45-degree dished sheets between the layers of elastomer. The radial bearing was made with alternate concentric steel sleeves separating the elastomer. Bench tests were conducted on the bearings to define their physical characteristics at -65°, +72°, and +160°F. These test data, presented herein, show significant stiffness changes at low temperature; however, indications are that with proper design, the bearings would be acceptable for tail rotor use throughout the temperature range.

Two blade grip bearing arrangements were whirl tested. One arrangement used a single chevron thrust bearing to carry both the centrifugal force and bending loads; the second used the 45-degree dished thrust bearing in conjunction with the radial bearing mentioned above. The tests showed acceptable frequency placement for the radial-thrust bearing arrangement. The chevron arrangement frequency placement was unsatisfactory.

Flight tests of the radial-thrust bearing arrangement showed acceptable blade and hub structural loads, which were in most cases comparable to those of the UH-1 tail rotor. Tail rotor mast and control oscillatory loads were found to be higher than those of the UH-1. Although difficult to define precisely due to differences in the test aircraft, the radial-thrust elastomeric bearing arrangement, in general, appears to be comparable to the elastomeric-Teflon journal bearing configurations tested previously. As in the previous program, the radial (flapping hinge) elastomeric bearing proved to be satisfactory.

As part of a Bell Helicopter Research Program, a brief flight evaluation of the single bearing arrangement was made. This tail rotor configuration, in combination with the blade control geometry used, was found to have unsatisfactory dynamic characteristics which precluded further testing.

It is concluded that the feasibility of the elastomeric bearing tail rotor has been proved.

FOREWORD

This report is submitted in compliance with provisions of United States Army Aviation Materiel Laboratories (USAAVLABS) Contract Number DAAJ02-67-C-0076, "Improved Concept of Elastomeric Bearings for UH-1 Tail Rotor Assembly." The work conducted during this program commenced upon receipt of the contract on 7 June 1967.

The program was conducted under the technical cognizance of Mr. E. R. Givens of the Aircraft Systems and Equipment Division of USAAVLABS. Principal Bell Helicopter Company personnel associated with the program were Messrs. W. Cresap, C. Fagan, P. Gibson, R. Lynn, G. Rodriquez, and J. White. Also, the writer wishes to acknowledge the technical assistance of Mr. R. Peterson and Mr. R. Nicoll of the Lord Manufacturing Company.

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INTRODUCTION

Bearings made with metal-elastomer laminations for use in oscillatory motion applications were investigated in 1961 by the Franklin Institute at Philadelphia, Pennsylvania. Under Air Force contract, they conducted an analytical and laboratory study which indicated the potential of the concept. Results of this early effort are reported in Reference 1. In early 1964, Bell Helicopter Company investigated the bonded-type elastomeric bearing further. This effort was principally concerned with defining the bearing characteristics and developing the fabrication techniques. The results of this work indicated that the bonded-type bearings would be suitable for helicopter application.

In 1965, Bell was awarded a contract by the United States Army Aviation Materiel Laboratories to investigate elastomeric bearings for application in a see-saw type tail rotor. This work included flight tests of two elastomeric bearing UH-1 experimental tail rotors, one bonded and one molded. With these configurations, the thrust bearings were used to carry the blade centrifugal force, and Teflon journal bearings were used to transfer the blade bending loads. The molded bearing flight test results were most promising, and a Bell IR&D program was initiated to evaluate the applications of the molded thrust bearing to various blade grip bearing arrangements. The flight test results from both programs are reported in Reference 2.

Additional investigations of new elastomeric bearings and tail rotor applications were required as a part of the subject program. The requirements of this program, which began in June 1967, included the design, fabrication, and test of new bearings for the rotor blade grips, and whirl test of two tail rotor configurations. Based on the initial test results, the most promising tail rotor configuration was to be flight tested throughout the speed range of the test helicopter. The results of these investigations are reported herein.

BEARING DESIGN AND FABRICATION

One radial and two thrust bearings were designed and fabricated for use in this program. All bearings were fabricated by the hot mold transfer method, which consists of stacking metal separator sheets in a steel mold, forcing hot elastomer into the mold, and maintaining pressure until the curing process is complete. The elastomer used in the new bearings is BTR-4 polybutediene. The radial bearing used in the tail rotor flapping hinge was fabricated using natural rubber.

FLAPPING HINGE RADIAL BEARING

The molded radial bearing, shown in Figure 1a, was used in the tail rotor flapping hinge. This bearing was designed during the previous program, and sufficient tests were conducted at that time to show the bearing to be satisfactory for the application and to establish its physical characteristics. Therefore, no further research was conducted on the elastomeric flapping bearing. One set of radial flapping bearings was used in the tail rotor throughout the program. The bearings were inspected at the end of the program, and no evidence of deterioration was visible.

CHEVRON THRUST BEARING

The chevron thrust bearing, shown in Figure 1b, was designed for use in the tail rotor blade grip. In this application (shown in Figure 2a), the bearing is required to allow the blade pitch change motions, to carry the blade centrifugal force, and to transfer the blade bending loads to the rotor yoke.

Two major design requirements, the torsional spring rate and the loaded dynamic bending spring rate, established the size of the chevron bearings. A torsional spring rate of 25 inch-pounds per degree (per blade) was established during the previous program as the maximum for acceptable rudder control pedal forces. Also, from previous work a dynamic blade to yoke bending spring rate of 12,000 inch-pounds per degree was estimated as the stiffness required to prevent pitch horn translation with changes in pitch link loads. This stiffness requirement was believed to be twice that of a previously tested 45-degree dished separator sheet thrust bearing. Therefore, chevron separator sheets and increased compression load area were used in an effort to achieve the required stiffness. With the increase in bearing cross-sectional area, an increase in elastomer pad stack was necessary to maintain the low torsional spring rate. Thus, a compromise in the desired shear spring rate and size resulted in a bearing of 1.50 inches inside diameter, 2.80 inches outside diameter, and 2.22 inches elastomer pad stack height. The elastomer stack

consists of 46 layers of elastomer 0.025 inch thick separated by formed steel sheets. The 45-degree thrust bearing tested had an inside diameter of 1.75 inches, an outside diameter of 2.63 inches, and an elastomer pad stack height of 1.75 inches.

45-DEGREE THRUST BEARING

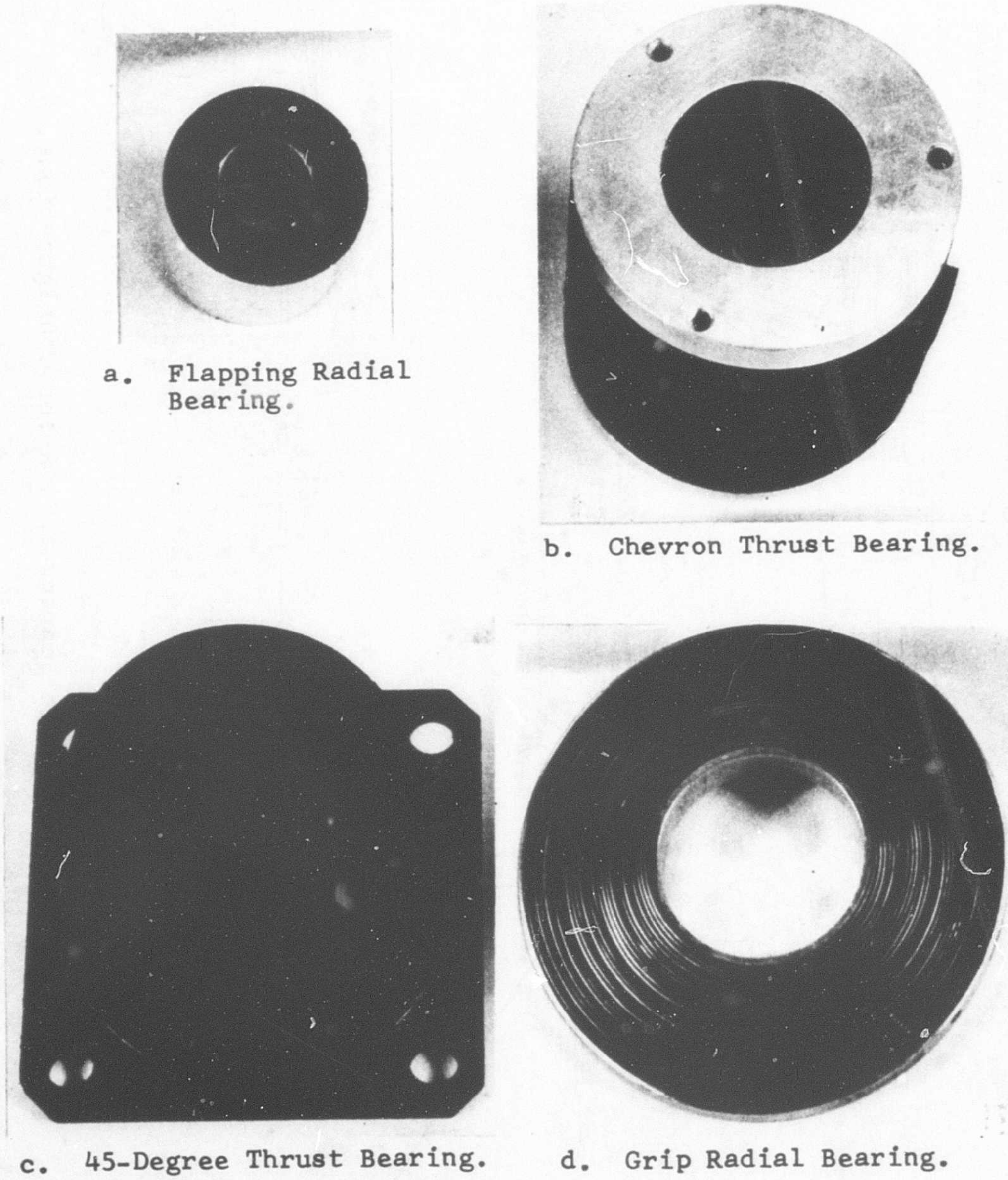
The 45-degree dished separator sheet thrust bearing, shown in Figure 1c, was designed for use in the rotor blade grip in combination with an elastomeric radial bearing. The design requirements for this bearing were to provide blade retention and, in conjunction with an inboard radial bearing, to allow the blade pitch change motions and transfer of the blade bending loads to the rotor yoke. With this design, shown in Figure 2b, the radial bearing provides a pivot and shear reaction for the blade bending loads.

The primary design requirement, which limited the thrust bearing's size, was a maximum torsional spring rate of 12.5 inch-pounds per degree. Since both the radial and thrust bearings are rotated angularly with blade motion, the sum of their torsional spring rates must not exceed the maximum of 25 inch-pounds per degree, as noted in the chevron bearing's section. The dynamic blade to yoke bending spring rate (of the radial-thrust bearing configuration) that was expected to provide an acceptable natural frequency placement for the hub was 8,000 inch-pounds per degree. Also, the compression load was a design factor in determining the minimum cross-sectional area of the thrust bearing. From previous static and endurance tests, 6,500 pounds per square inch was estimated as the allowable load for satisfactory bearing service in this type of application. The design compression load used (created by blade centrifugal force) was 20,000 pounds. Thus, the 45-degree thrust bearing envelope, which accommodates the above requirements, consisted of a cylindrical shape of 1.44 inches inside diameter, 2.50 inches outside diameter, and 1.80 inches elastomer pad length. The elastomer stack for this bearing consists of 38 layers of elastomer 0.025 inch thick separated by 45-degree dished separator sheets.

GRIP RADIAL BEARING

The radial bearing, shown in Figure 1d, was designed to support the blade root loads when assembled with the thrust bearing as shown in Figure 2b. The major design requirements were to satisfy the torsional spring rate and the oscillatory radial load. The torsional spring rate limit, as discussed in the previous paragraph, was 12.5 inch-pounds per degree. Also, the bearing was designed to withstand an oscillatory radial load of ± 2,000 pounds at a frequency of 1,650 cycles per minute. The radial oscillatory load and torsional spring rate requirements resulted in a bearing with 29 concentric circular layers of

elastomer separated by concentric steel separator rings, each 0.015 inch thick. To achieve the required low torsional spring rate, the bearing was tapered from 0.32 inch wide at the outer diameter to 0.50 inch wide at the inside diameter.



a. Flapping Radial Bearing.

b. Chevron Thrust Bearing.

c. 45-Degree Thrust Bearing.

d. Grip Radial Bearing.

Figure 1. Elastomeric Bearing Configurations.

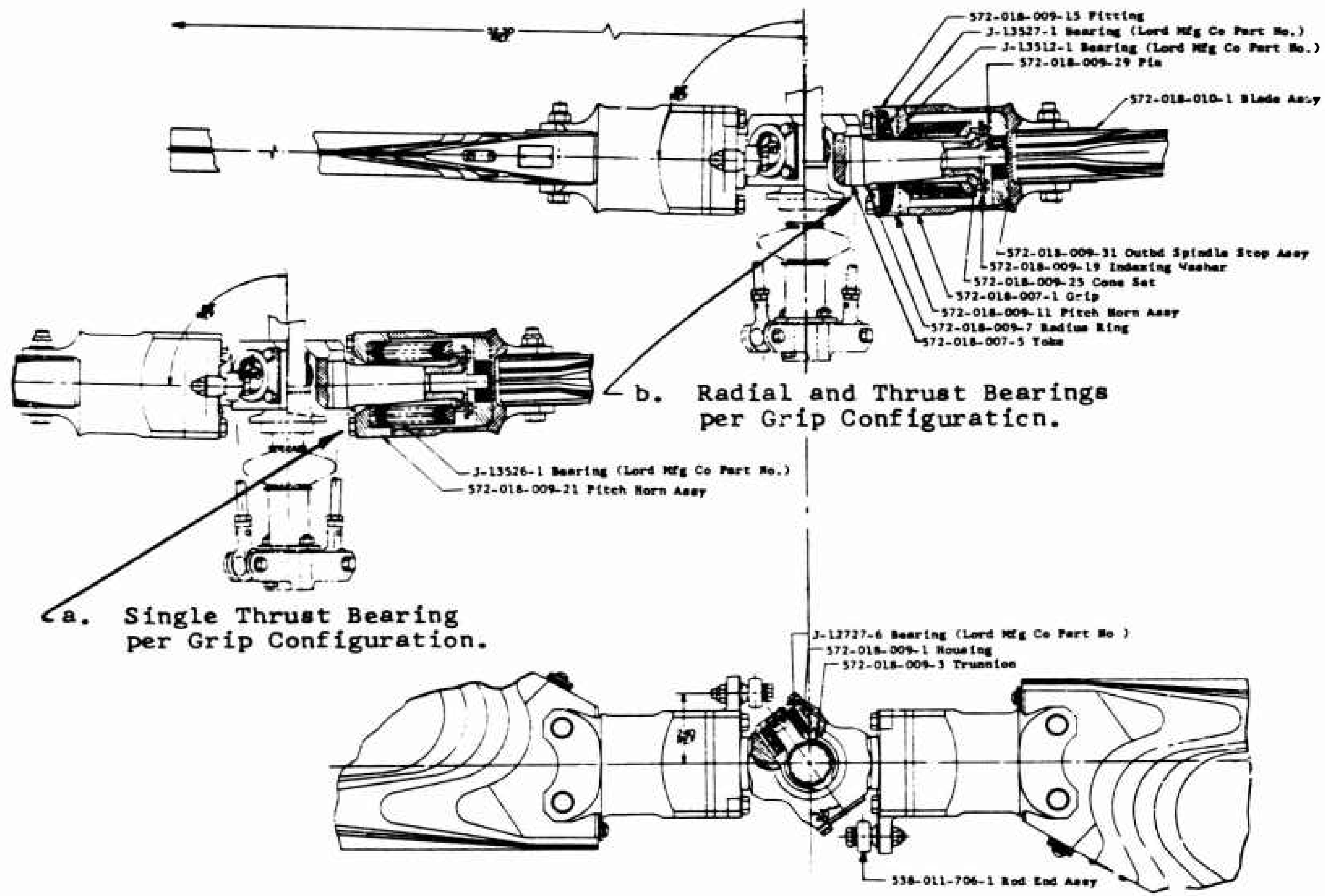


Figure 2. Elastomeric Bearing Tail Rotor Configurations.

BEARING BENCH TESTS

The radial grip, 45-degree thrust, and chevron thrust bearings were bench tested to determine their load-carrying ability, spring rates, and damping characteristics. The test results are discussed in the following paragraphs.

ANGULAR DEFLECTION

Angular deflection (of the thrust bearing end) versus bending load data were determined for the two thrust bearings using the apparatus sketched in Figure 3. These bending data are presented in Figure 4 for the chevron and the 45-degree thrust bearings (while under a compression load of 20,000 pounds). Angular deflection versus torsional load data (at temperatures of -65°, 72°, and 160°F), including hysteresis loops, are shown for these bearings in Figures 5 and 6. The torsional data were obtained using the same apparatus as that sketched in Figure 3. However, the load was applied in a direction tangent to the bearings, thus applying a torsional load to the bearings. Torsional hysteresis loops were also acquired for the radial bearing using the apparatus shown in Figure 7, and the test results are presented in Figure 8 for temperatures of -65° and 72°F.

COMPRESSION DEFLECTION

The 45-degree thrust and chevron thrust bearings were loaded in compression, and their spring rates are graphically presented in Figure 9 for temperatures of -65°, 72°, and 160°F. In addition, the radial bearing was loaded radially as shown in Figure 7. The radial bearings for both bench and flight tests were loaded in this manner, and the results are compared in Figure 10. The difference in the two curves is attributed to a change in elastomer stack type and stiffness.

LATERAL SHEAR DEFLECTION

The 45-degree thrust and chevron thrust bearings were subjected to shear loads (while under 16,000- and 20,000-pound compression loads and at room temperature) as shown in Figure 11. These shear test results appear in Figure 12.

DISCUSSION OF RESULTS

The bearing characteristics determined during bench tests compared very well with the calculated data except for the torsional spring rate of the grip radial bearing and the lateral stiffness of the chevron bearing.

Figure 8 shows that a torsional load of 300 inch-pounds is required for a maximum angular rotation of 16 degrees. This is an average of 19 inch-pounds per degree. (The desired spring

rate was 13 inch-pounds per degree). The increased torsional stiffness was not a problem for hot and normal temperatures, since the 45-degree thrust bearing used in conjunction with this bearing was 11 inch-pounds per degree. The total of 30 inch-pounds per degree for both bearings increases to approximately 60 inch-pounds per degree at a temperature of -65°F. This low-temperature spring rate is unacceptable for hydraulic boost-off helicopter operation. However, bearing motion during normal helicopter operation causes an increase in bearing temperature (hysteresis loss) which reduces the stiffness. It is believed (from unpublished data) that the low-temperature bearing spring rate will be reduced sufficiently for satisfactory hydraulic boost-off operation after a normal 5-minute helicopter ground warm-up operation (1.25 times room-temperature spring rate).

Figure 4 shows the face bending of the chevron thrust bearing to be considerably greater than that of the 45-degree thrust bearing. However, the shear data plotted in Figure 12 indicate that the lateral shear stiffness of the chevron thrust bearing rotor configuration may be insufficient to prevent pitch horn translation when the pitch link loads are applied.

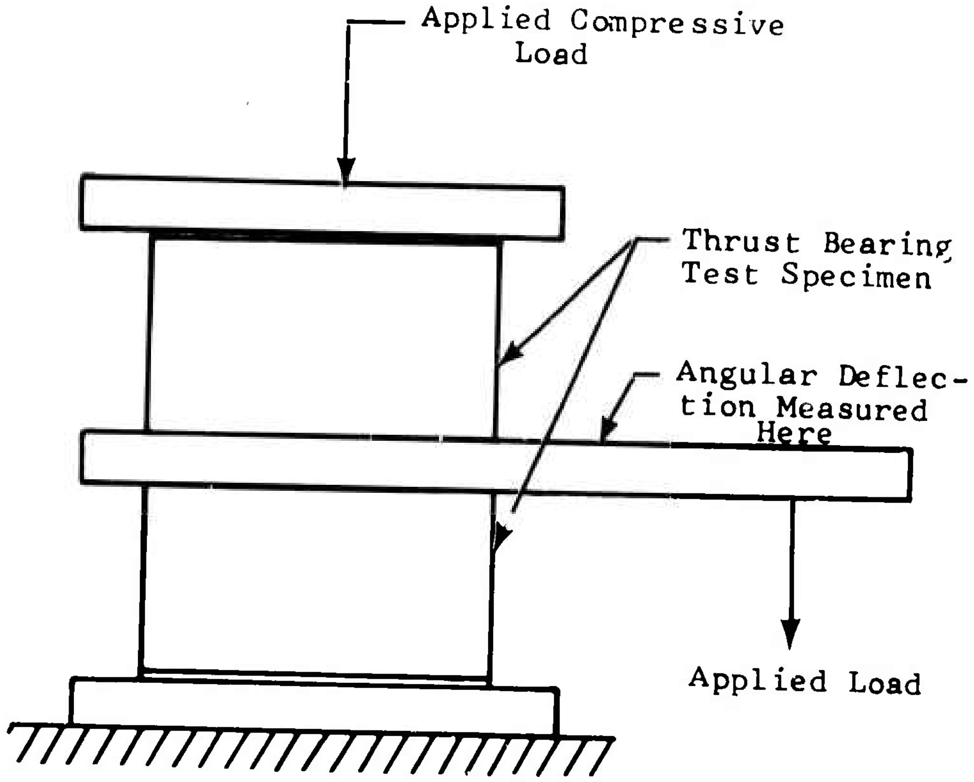


Figure 3. Schematic Diagram of Thrust Elastomeric Bearing Bending Test Apparatus.

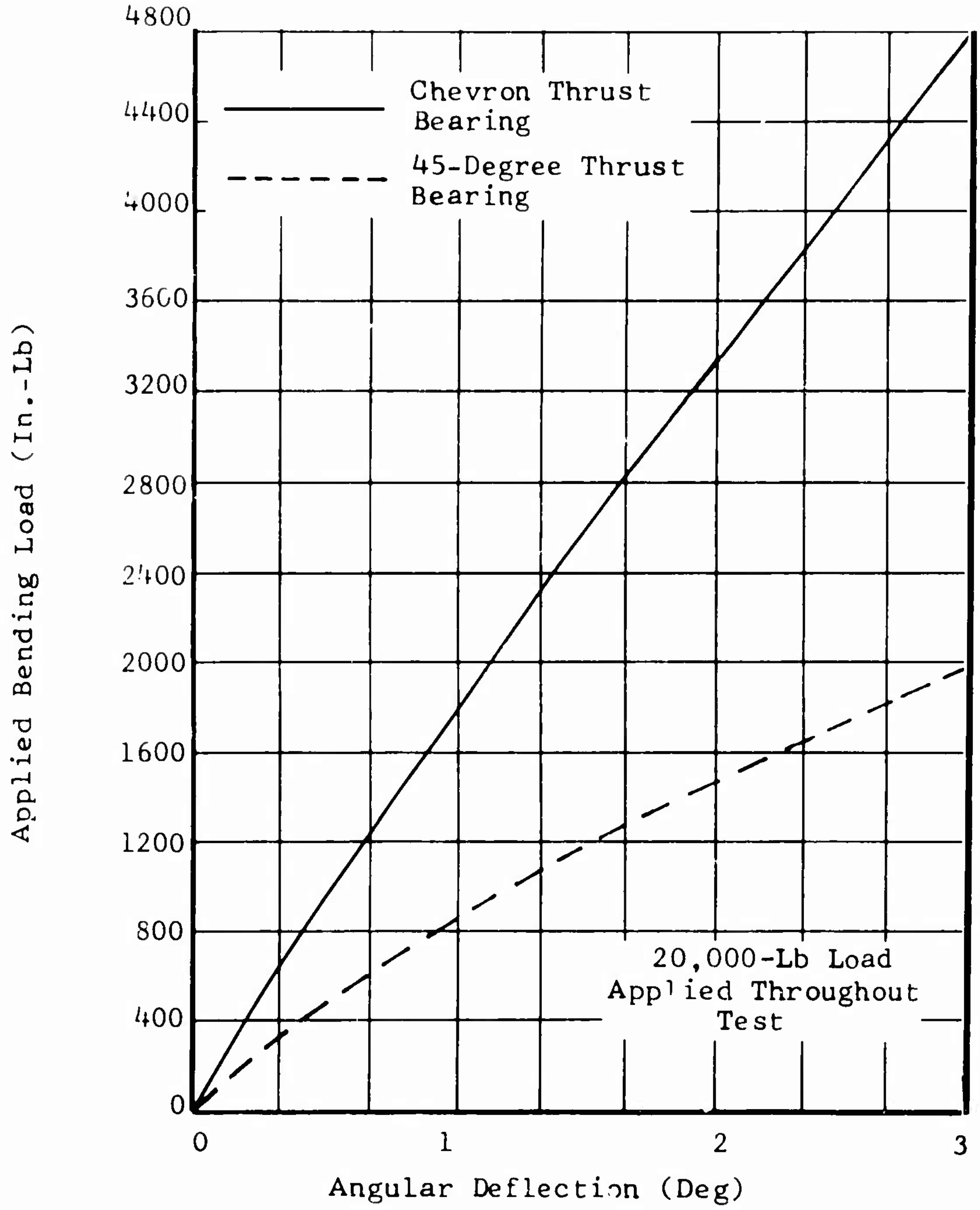


Figure 4. Bending Spring Rate of the Chevron and 45-Degree Thrust Elastomeric Bearings.

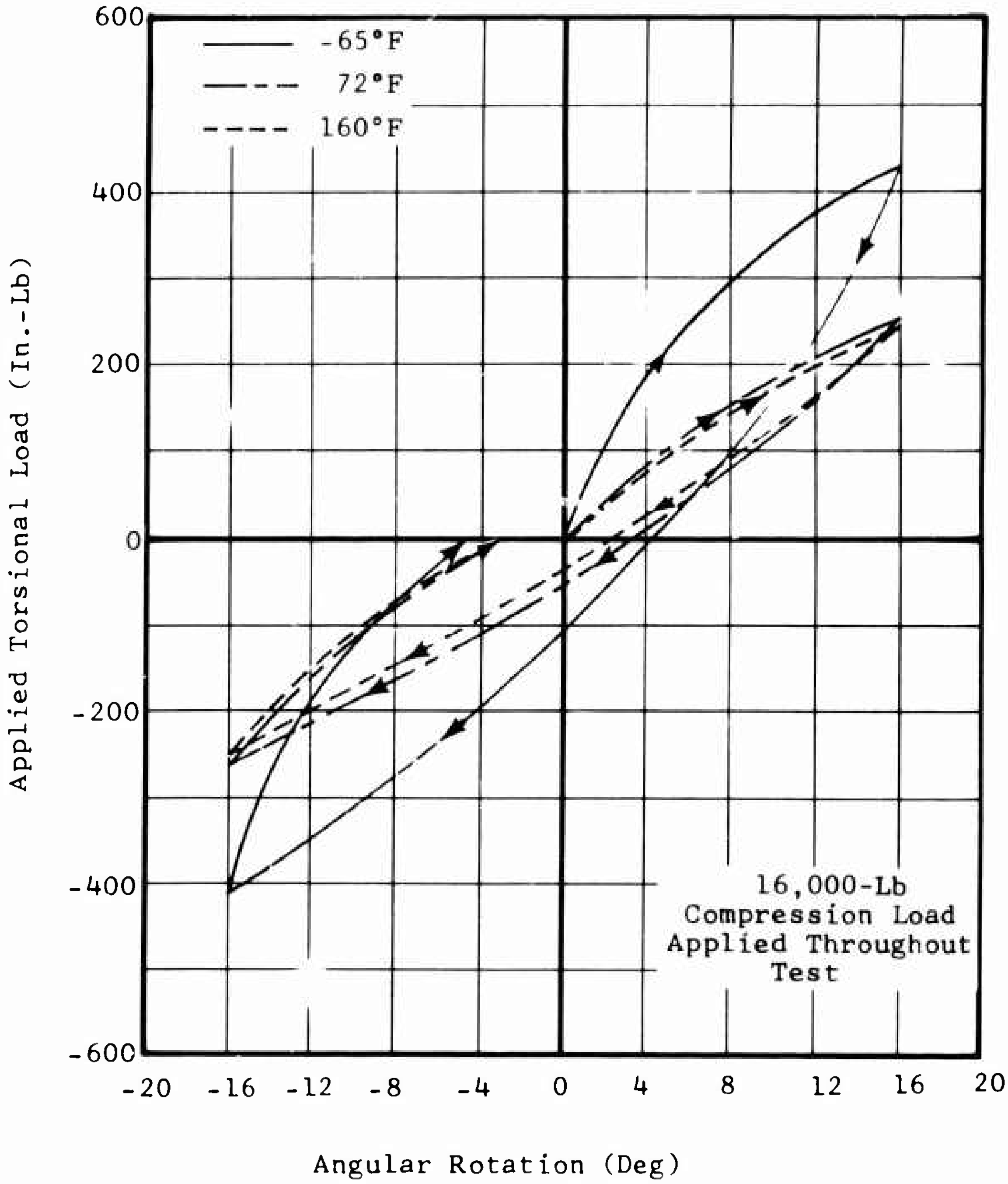


Figure 5. Torsional Spring Rate and Hysteresis for the Chevron Thrust Elastomeric Bearing.

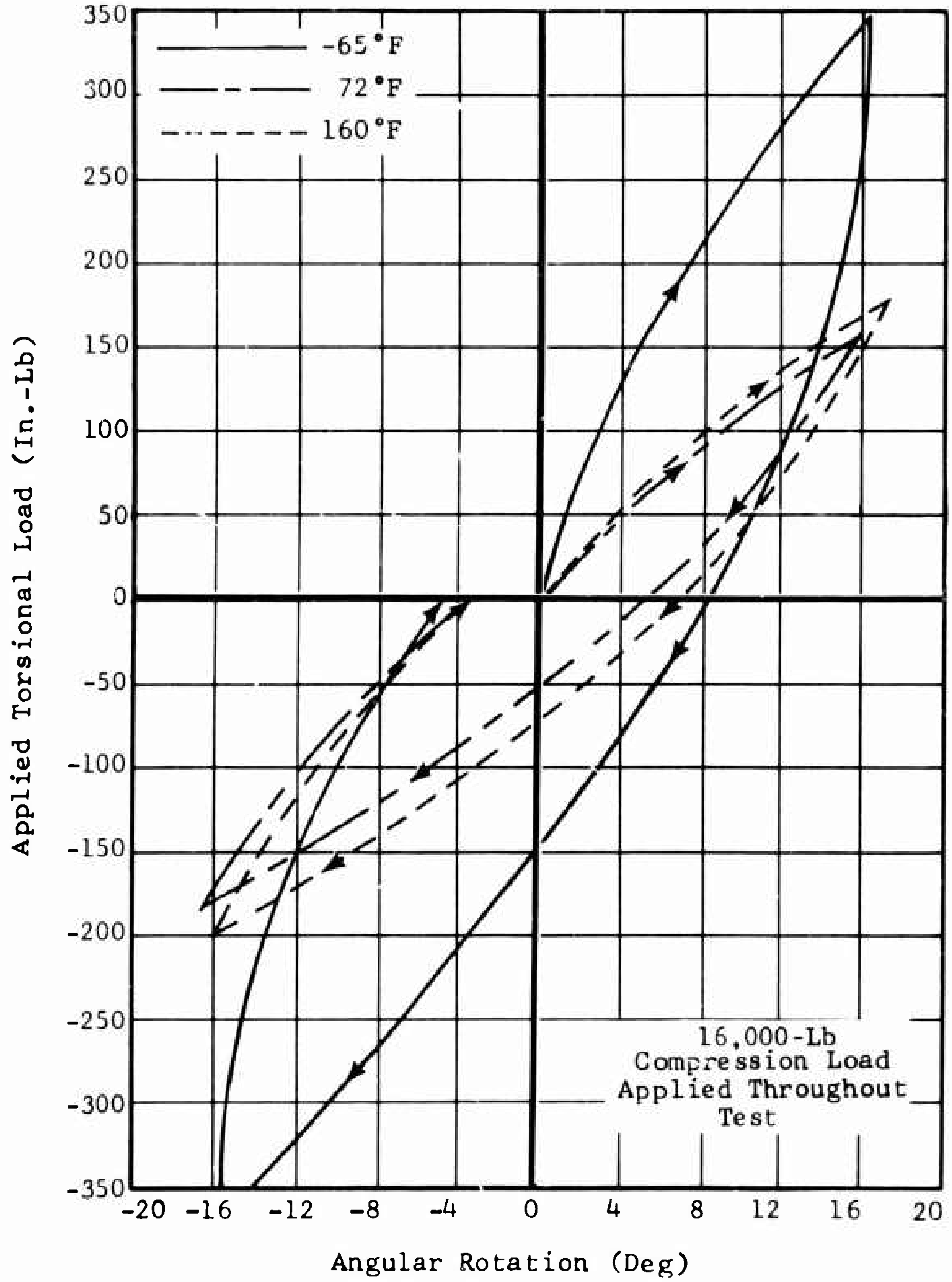


Figure 6. Torsional Spring Rate and Hysteresis for the 45-Degree Thrust Elastomeric Bearing.

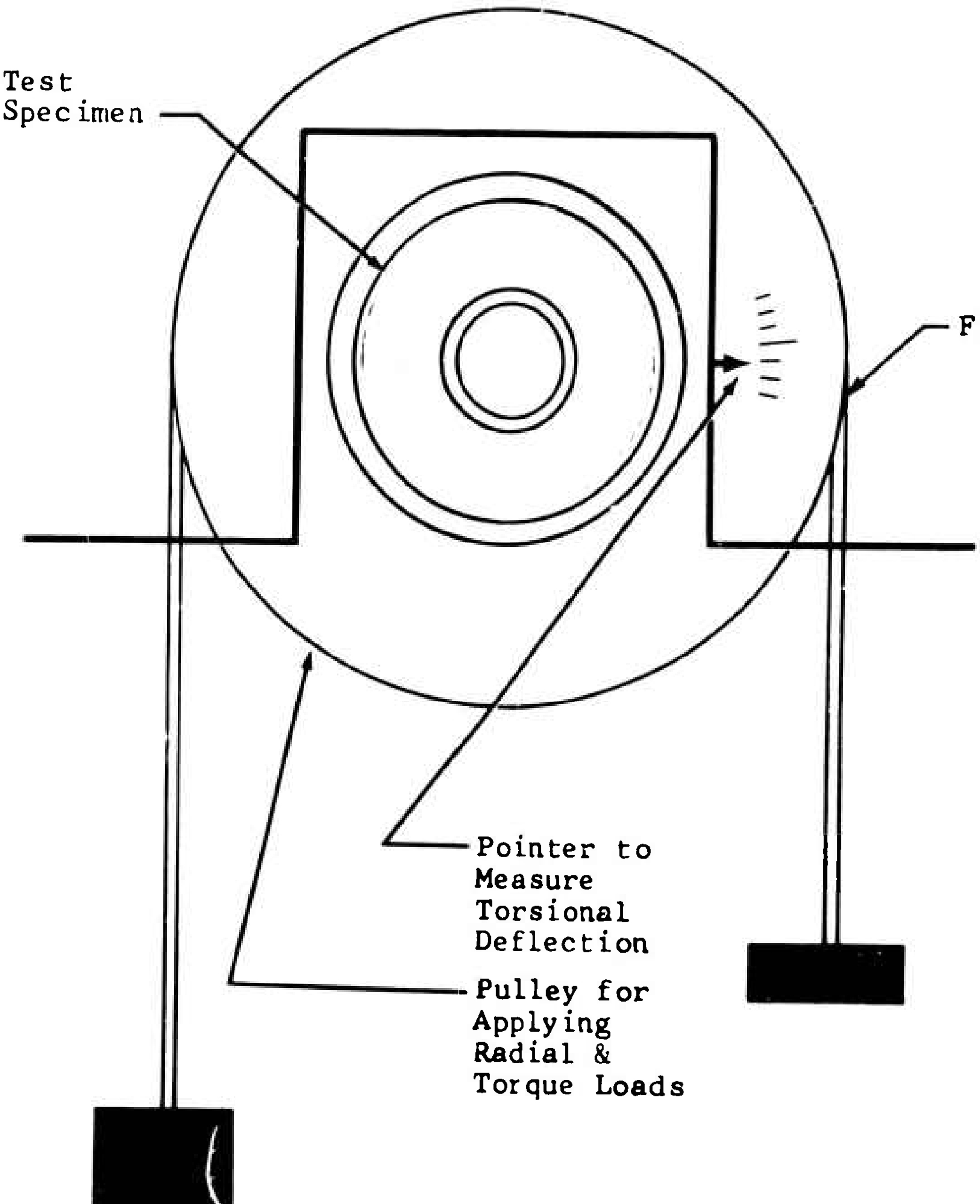


Figure 7. Schematic Diagram of Test Setup for Torsional and Radial Loading Tests of the Radial Grip Bearing.

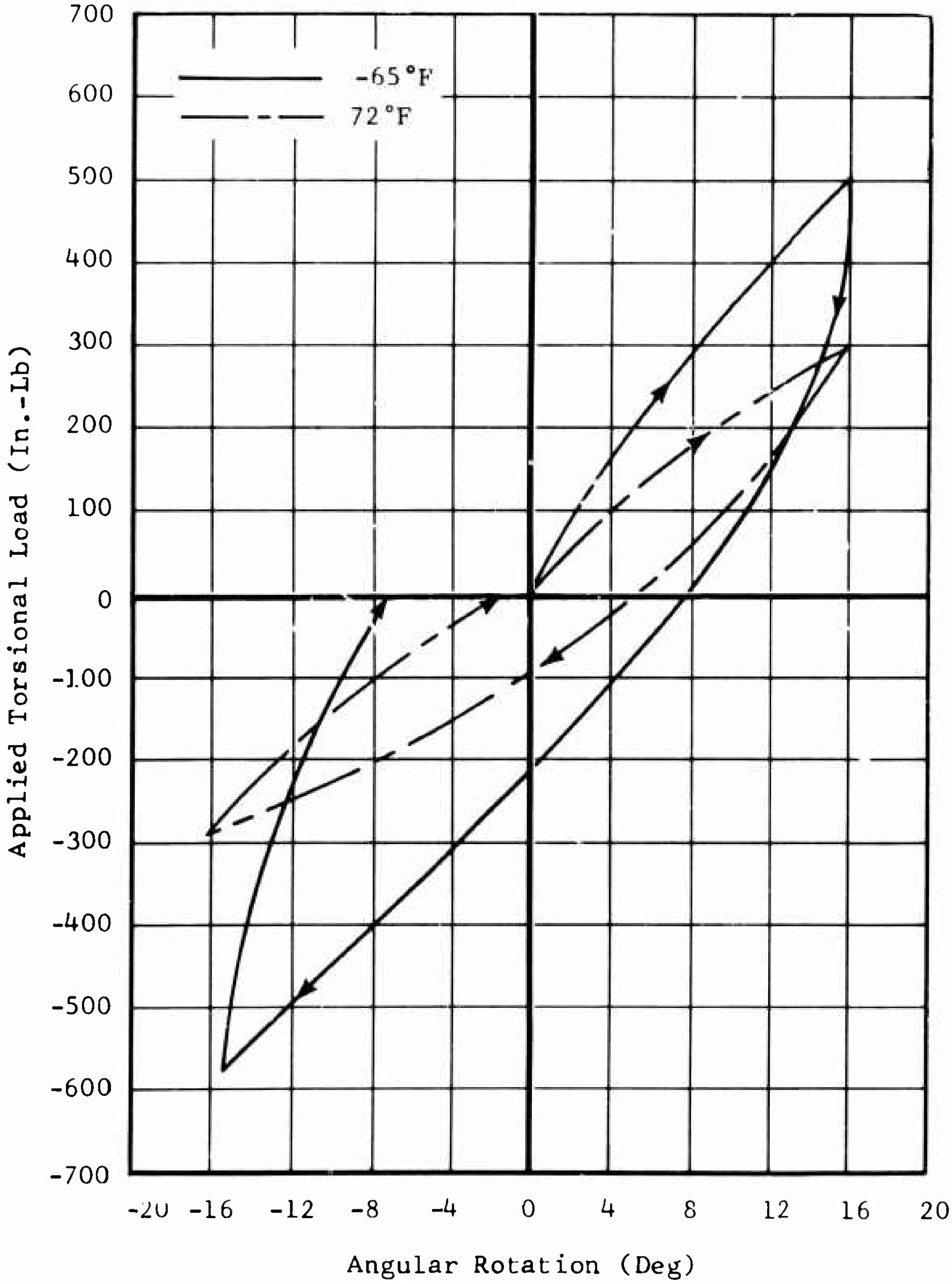


Figure 8. Torsional Spring Rate and Hysteresis for Radial Elastomeric Bearing.

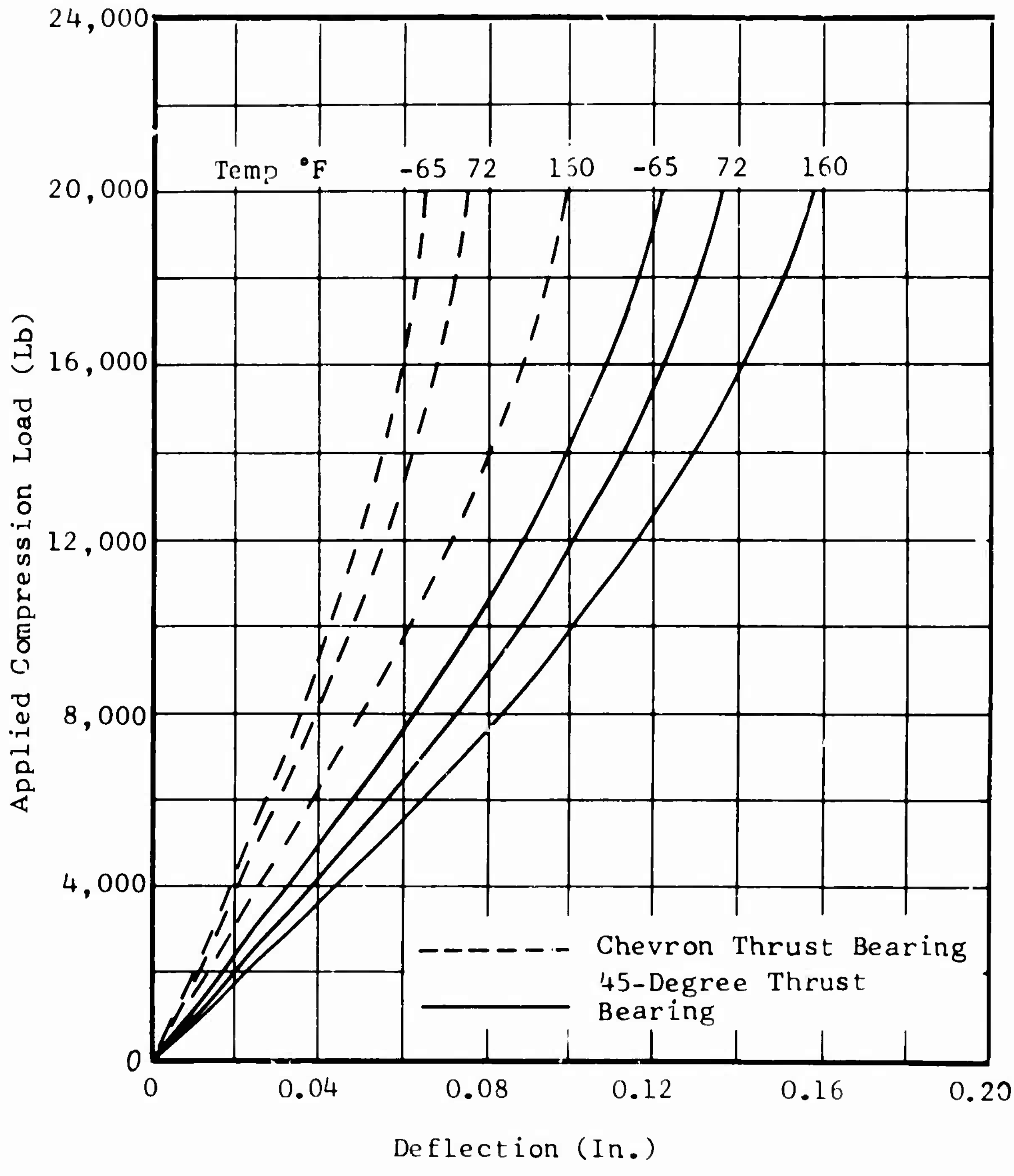


Figure 9. Compression Spring Rate of the Chevron and 45-Degree Thrust Elastomeric Bearings.

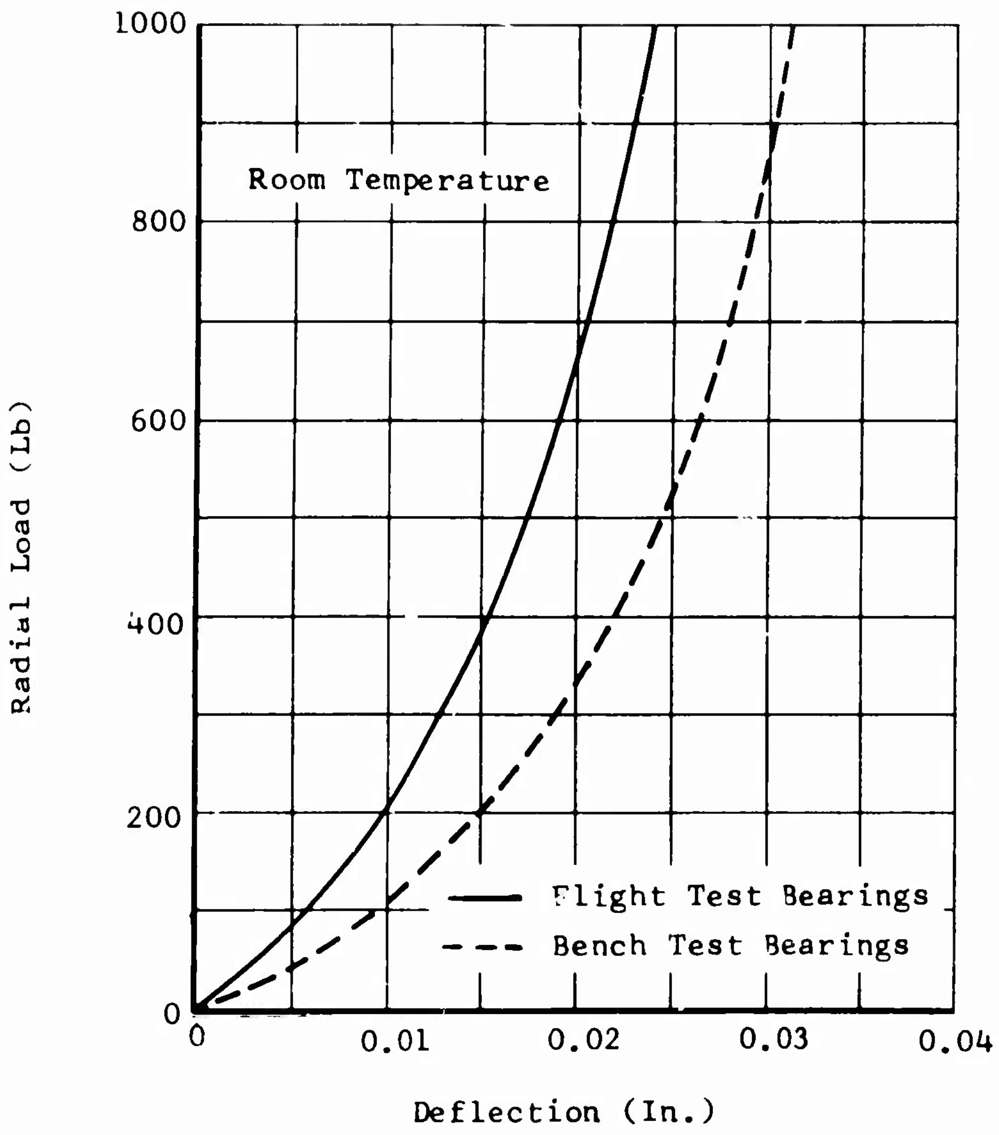


Figure 10. Radial Spring Rate of the Radial Elastomeric Bearing.

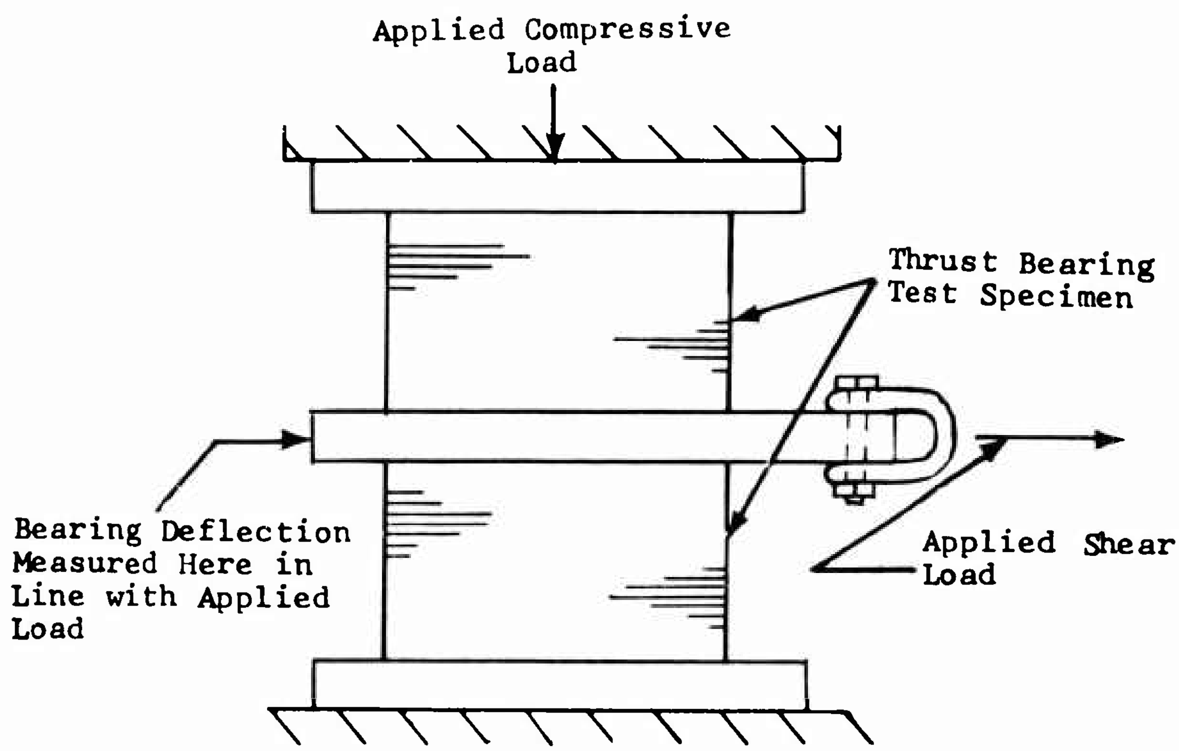


Figure 11. Schematic Diagram of Elastomeric Thrust Bearing Shear Test Apparatus.

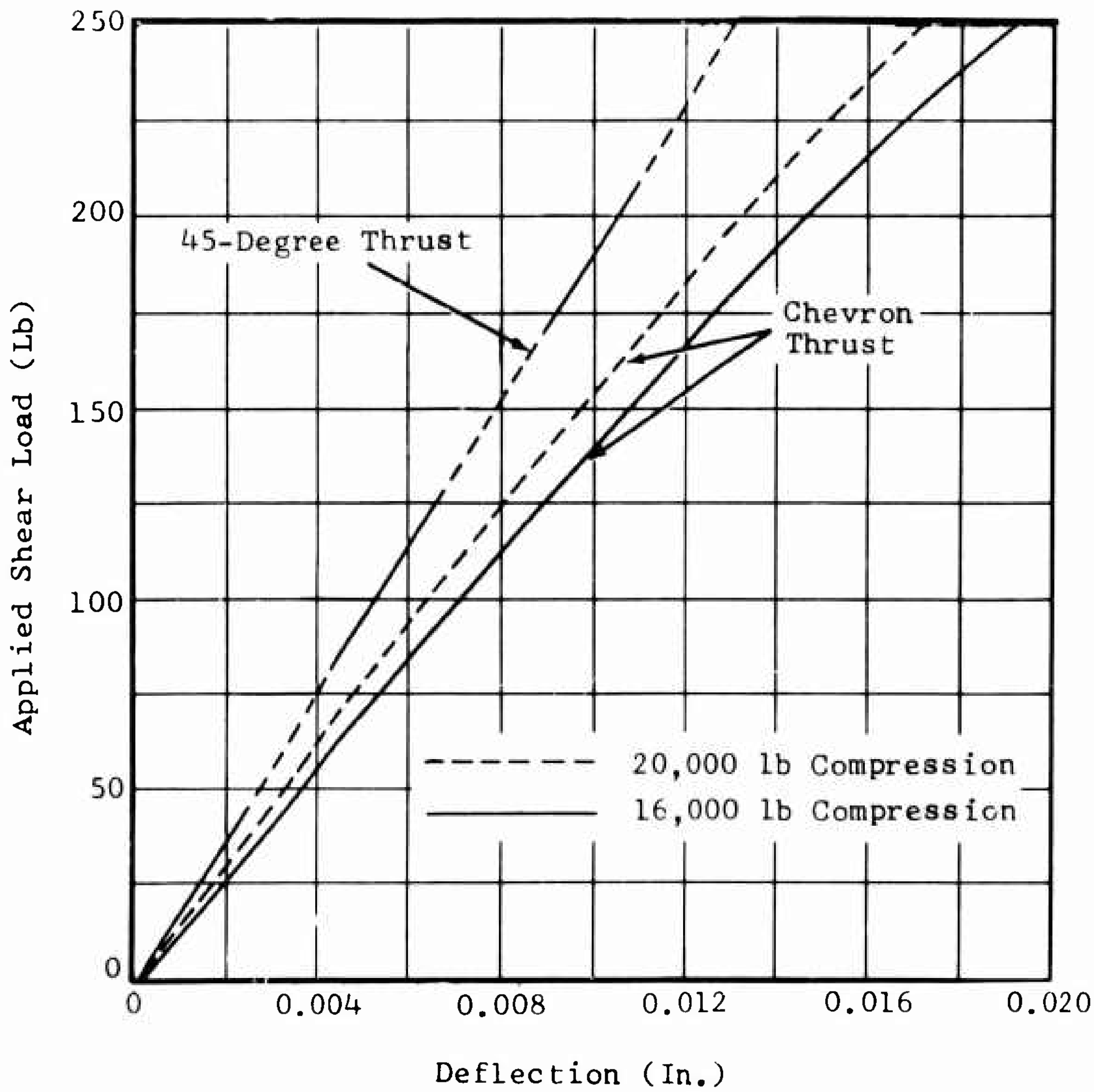


Figure 12. Shear Spring Rate of the Chevron and 45-Degree Thrust Elastomeric Bearings.

WHIRL TESTS

The two experimental tail rotor configurations shown in Figures 2a and 2b were whirl tested. The purposes of these tests were to investigate the rotor natural frequencies of each and to demonstrate their airworthiness. Both rotors were balanced and tracked prior to the tests. The tests were conducted using a 100-horsepower stationary whirl test machine, located at the Bell facility. The test equipment has provisions for automatically changing the blade pitch and rotor rpm during the whirling operation.

TEST CONDITIONS

Rotational dynamic frequency data were acquired by sweeping the rotor speed from 200 to 1,650 rpm while maintaining constant blade pitch settings of 0, +4, +8, and +12 degrees.

The airworthiness evaluation consisted of 6 hours of continuous endurance testing for each configuration. The test conditions consisted of rotating the rotor at 1,650 rpm and cycling the blade pitch from -5 degrees to +12 degrees and back, at a frequency of nine times per minute. The loads and motions imposed on the bearings during these tests are believed to be sufficient to establish the structural airworthiness of the rotor for test purposes.

WHIRL TEST RESULTS

Whirl tests of the elastomeric bearing tail rotor configurations were completed without difficulty; no bearing damage or excessive loads were encountered. The load data taken at the beginning of each airworthiness test were compared with those taken at the end of the test, and only minor differences were noted.

The pitch link loads for both configurations are presented in Figure 13 along with standard UH-1 tail rotor whirl test data. The comparison shows that the pitch link load changes for the elastomeric bearing rotors are approximately equal to those of the standard rotor for the same change in blade pitch. Therefore, the rudder pedal control forces for the experimental rotors were expected to be satisfactory.

The rotor natural frequency data acquired during the whirl tests are discussed in the Tail Rotor Dynamics section of this report (see page 23).

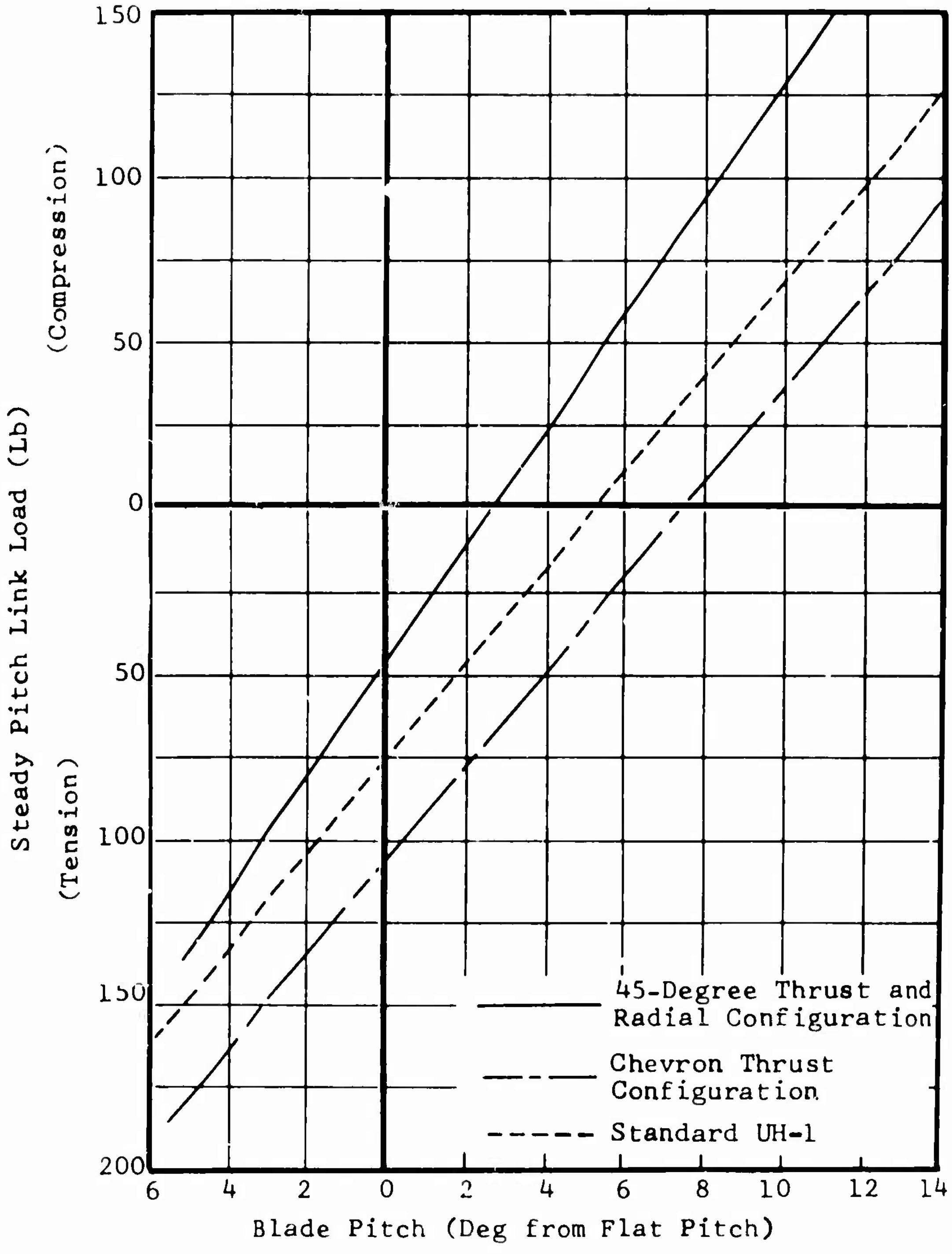
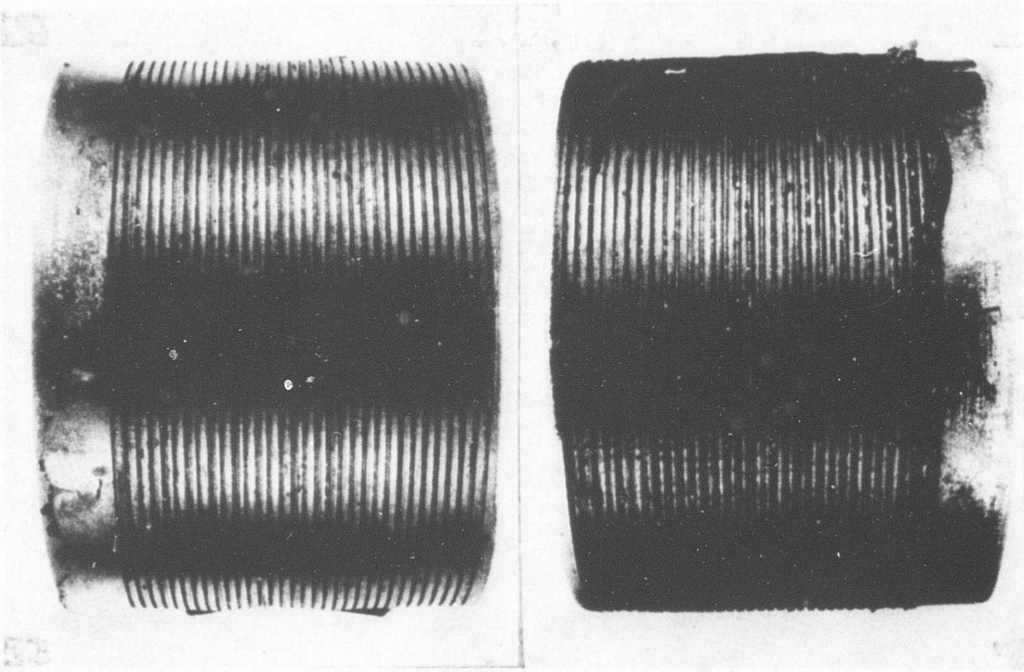


Figure 13. Whirl Stand Tail Rotor Steady Pitch Link Loads (at 1650 RPM During Pitch Change Cycle).

ACCELERATED ENDURANCE TESTS

During this program, Bell Helicopter Company also conducted independent research and development elastomeric bearing tests. The purposes were to investigate the blade load path through the thrust bearing and to estimate the bearing's service life in the tail rotor application. The test rotor bearing configuration used was similar to the radial-thrust bearing arrangement tested under this contract. However, a Teflon journal bearing was used in the blade grip instead of the elastomeric radial bearing used in the contracted program. The test conditions selected for the accelerated endurance tests were such that 10 hours of testing were expected to be representative of 1 year of normal service life. Figure 14 shows the condition of the bearing before and after the endurance tests and tabulates the test conditions and results.



BEFORE TEST AFTER TEST

THE ABOVE PHOTOGRAPHS SHOW THE ELASTOMERIC THRUST BEARING BEFORE AND AFTER AN ACCELERATED ENDURANCE WHIRL TEST REPRESENTATIVE OF 5 YEARS OF NORMAL SERVICE OPERATION IN THE EXPERIMENTAL UH-1 TAIL ROTOR.

BREAKDOWN OF THE TOTAL ENDURANCE TEST			
No. of Hours Rotating	Tail Rotor rpm	Number of Start-Stops	Number of Blade Pitch Cycles (-5° to +12° to -5°)
7.98	1750	129	2,570
46.65	1640	621	17,970
	Nonrotating		10,400
54.6		750	30,940

Figure 14. Endurance Test of 45-Degree Thrust Bearing

TAIL ROTOR DYNAMICS

The dynamics of the two elastomeric bearing tail rotor configurations were evaluated on the basis of frequency calculations and experimental test results.

ANALYTICAL

The natural frequencies of the two elastomeric bearing tail rotor configurations were computed from their respective flapwise and chordwise stiffness and mass distributions. The values used in the calculations are based on the rotors shown in Figures 2a and 2b using standard UH-1 blades without weights. The analytical results are shown graphically (Figures 15 and 16), in relation to the exciting frequencies, with rotor natural frequency data from the whirl and flight test programs. The calculated data in Figure 15a show the second flapwise collective mode to be near the four-per-rev exciting force. However, no corrective measures were taken, since past calculations of this mode have usually been found to be slightly high, owing to difficulty of representing hub stiffness, impedance, and boundary conditions. During the previous elastomeric bearing tail rotor program, the first flapwise cyclic mode was found to be excited by the operating three-per-rev forcing function. Thus, particular attention was given the calculated results for this cyclic mode. Figure 15b shows the blade S-ing mode to cross the three-per-rev excitation frequency in the rotor operating range. This mode is principally affected by blade stiffness and mass distribution and, to a lesser degree, by hub stiffness; therefore, a 1-pound blade mid-span weight was recommended to lower the mode.

EXPERIMENTAL

Whirl Stand

Natural frequencies were determined during whirl tests and helicopter ground run for both configurations. The intermediate resonant points recorded (during rpm sweeps at constant blade pitch) were plotted, and the resulting curves were extrapolated to operating speeds, as shown by Figures 15 and 16.

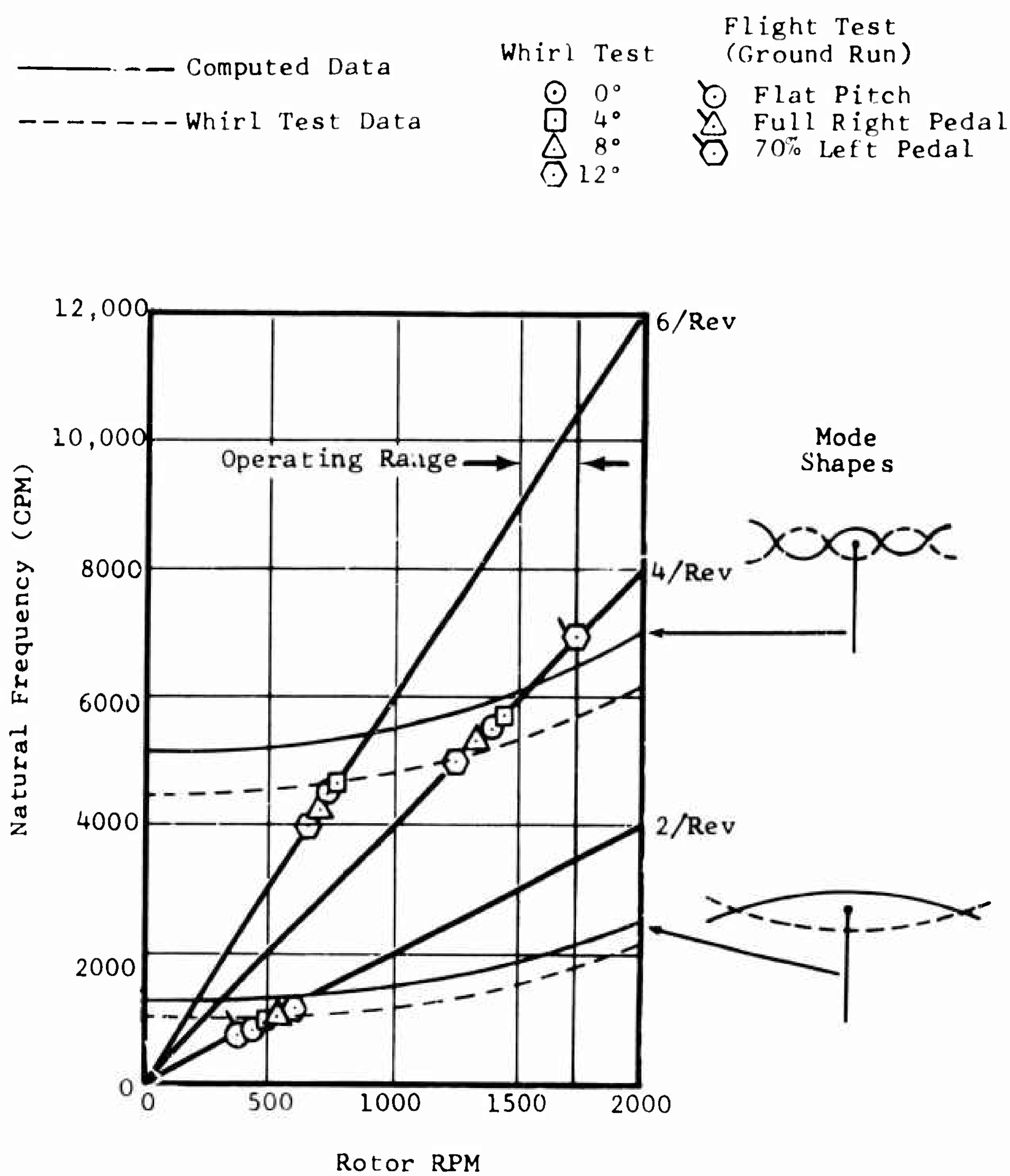
The whirl tests were conducted using standard UH-1 blades. The collective modes are shown to be well located with respect to the exciting forces for helicopter operating speeds. However, Figure 15 shows that three-per-rev resonance was present at helicopter operating speeds for the first cyclic flapwise mode. Since placement of this mode can be accomplished by blade mass changes, and 1-pound mid-span blade weights were installed at blade station 25 for the flight test program of the radial-thrust bearing configuration.

Helicopter Ground Run

The first rotor evaluated during the helicopter ground run was the radial-thrust bearing configuration using 1-pound mid-span weighted blades. For this rotor, the first flapwise cyclic mode is located below three-per-rev resonance, as can be seen in Figure 15b. The other cyclic modes, as well as the collective modes, are well located for this configuration, as shown by the plotted data.

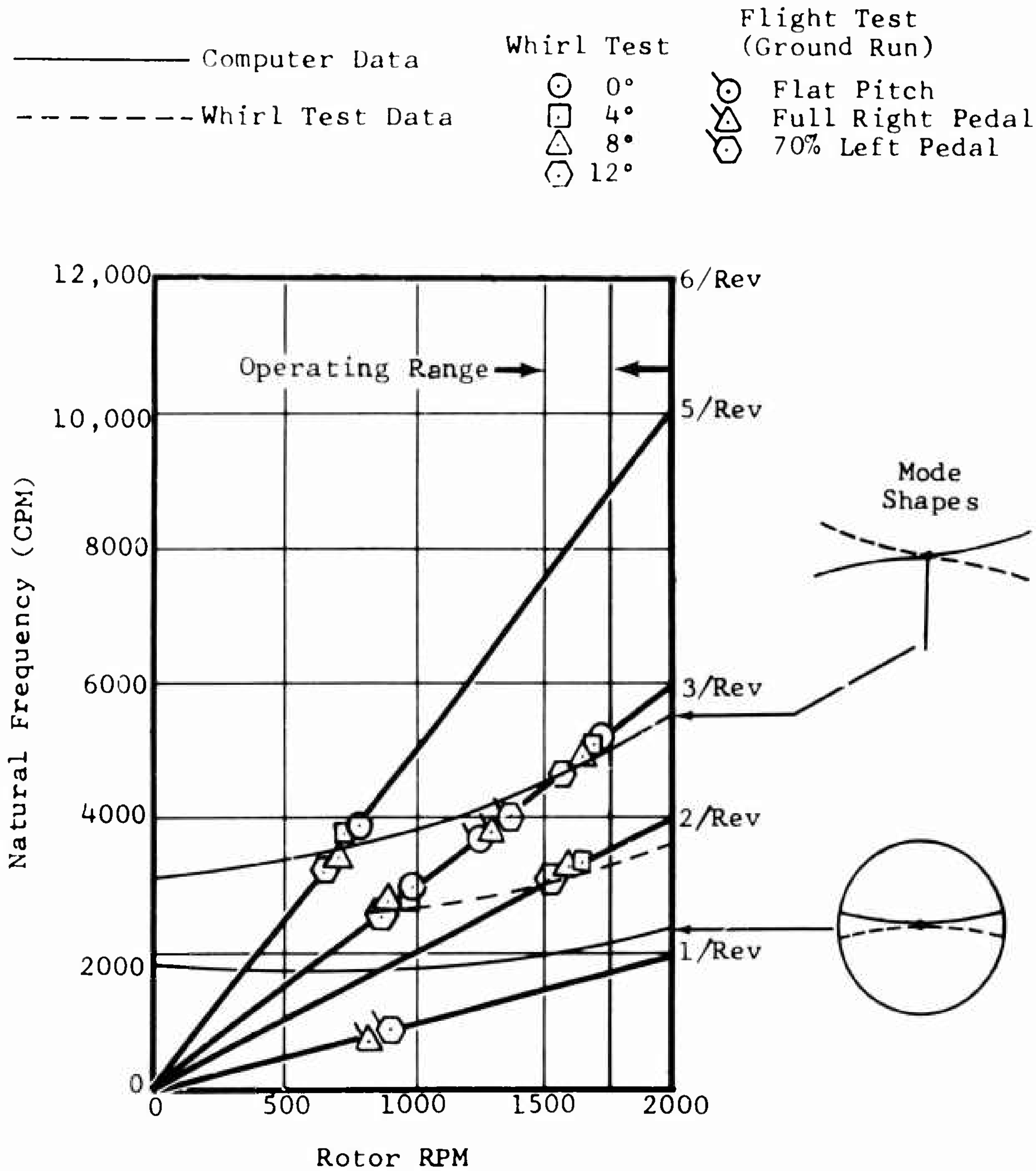
Similar dynamic investigations were made with the chevron bearing rotor configuration using standard UH-1 blades (no mid-span weights). The second flapwise collective mode is shown in Figure 16a at the rotor operating range; thus, an increase in in-flight four-per-rev yoke beam bending loads was expected (compared to the previous configuration). Also, at operating rotor speed, the first cyclic flapwise mode is located near the three-per-rev resonance as shown in Figure 16. The stiffness of this grip bearing configuration is discussed further in the Flight Test section.

The behavior of the first inplane symmetric mode for both configurations suggests a strong nonlinearity such that the blade acts as if it is hinged up to approximately 900 rpm and then stiffens (Figure 15b). Such a nonlinearity (which may be due to bottoming against one side of the spindle as the steady load increases) cannot be treated by the analysis used. The linear elastic element used in the hub region gives a first inplane frequency which lies between the soft and stiff nonlinear values. Annoying buzz or limit cycle vibrations may result from such a system.



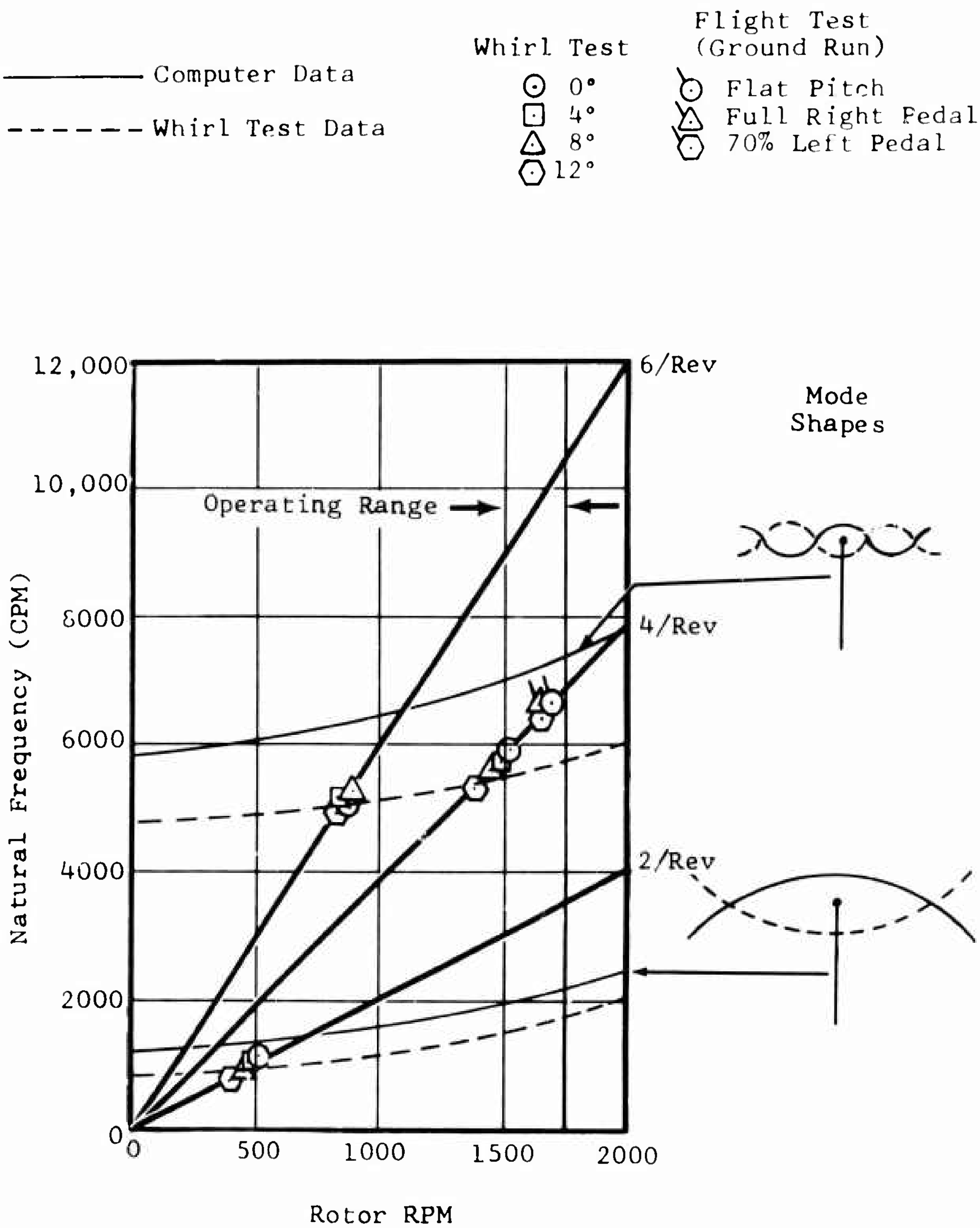
a. Collective Mode

Figure 15. Coupled Rotor Natural Frequencies for the Radial Plus 45-Degree Thrust Bearing Tail Rotor Configuration.



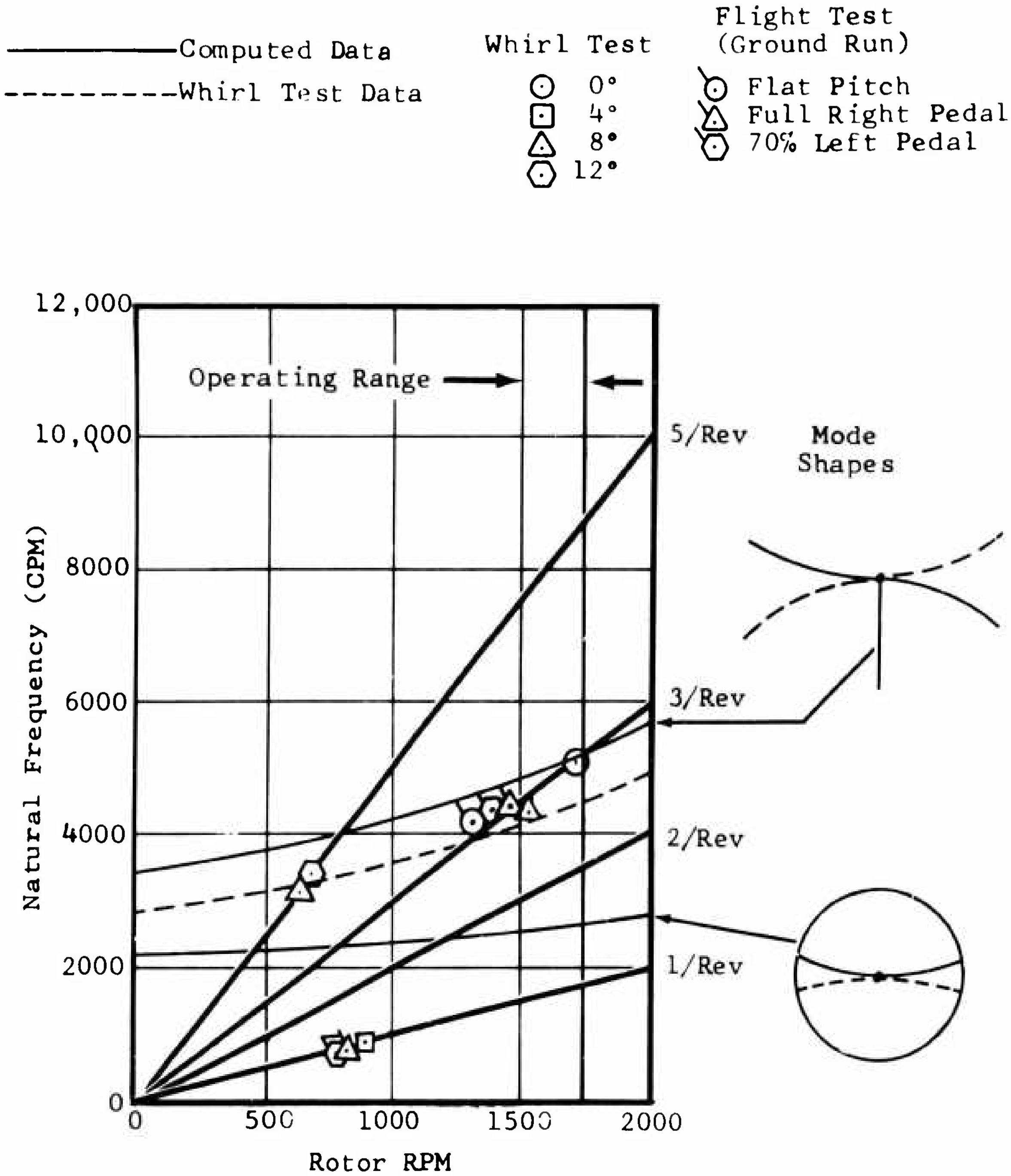
b. Cyclic Mode

Figure 15. Concluded.



a. Collective Mode

Figure 16. Coupled Rotor Natural Frequencies for the Chevron Thrust Bearing Tail Rotor Configuration.



b. Cyclic Mode

Figure 16. Concluded.

FLIGHT TEST

Two all-elastomeric bearing tail rotor configurations were flight tested. The primary configurations selected for flight test used a radial and thrust bearing in each blade grip to carry the blade loads. The other rotor configuration used a single thrust bearing per blade grip to accomplish this purpose. Both rotors used radial elastomeric bearings in the flapping hinge (same size and type used in the previous elastomeric bearing tail rotor program).

FLIGHT TEST EQUIPMENT

The two rotors flight tested are shown prior to assembly in Figure 17, and the radial-thrust bearing configuration is shown installed on the helicopter in Figure 18. A standard UH-1C helicopter (Serial Number 66-602) was used for the flight test program; however, a UH-1B helicopter was used during the Reference 2 flight test program. The significant component differences between these two aircraft are the main rotors and the tail fins. A complete description of the UH-1C helicopter is given in Reference 3.

The tail rotors were instrumented as shown by Figure 19. Also, both configurations were balanced prior to installation and tracked before each flight.

PRIMARY CONFIGURATION TEST RESULTS

Flight tests were conducted on the radial-thrust bearing configuration assembled as shown in Figure 2b. One-pound mid-span weights were added to the blades. The tests consisted of level flights to the power-limited forward speed of the helicopter, dives, climbs, autorotation, and sideward flights. The data presented herein were acquired with the helicopter gross weight at 7500 pounds and the center of gravity located at neutral. Since the main rotor and fin significantly affect the tail rotor loads, a detailed comparison of the results of this program and data from the previous elastomeric bearing tail rotor program cannot be made. Comparable data for the standard UH-1C tail rotor obtained from Reference 4 are presented graphically with level flight results from this program in Figures 20 through 25.

Tabulated flight test results for maneuvers and forward flight to 120 knots are presented in the Appendix. Test data were obtained during dives at 130 and 140 knots. The helicopter gross weight for these tests was 6500 pounds. The maximum loads, obtained at the lower gross weight, were of the same magnitude as the loads obtained during the maximum speeds of the 7500-pound gross weight.

DISCUSSION OF RESULTS

With all helicopter tail rotors, both chordwise and beamwise out-of-balance has an adverse effect on the rotor loads. Balance of the elastomeric bearing tail rotor is no more critical than for the standard UH-1 tail rotor; however, more caution must be exercised during the balance operation. Due to the softness of the elastomeric thrust bearing (under no compression load), difficulty is encountered in properly aligning the blade with the yoke spindle. The extent of out-of-balance for this rotor during flight test is not known. Specific flights were not conducted to evaluate the effects of balance, since the existence of this possible problem area was not discovered until the test vehicle was no longer available.

The rotor yoke and blade oscillatory loads and the shaft oscillatory torque are satisfactory. However, the blade chord loads are slightly higher than those of the standard rotor. The level-flight elastomeric bearing flapping motions (see Figure 25) are low when compared with the UH-1C data. The flapping reduction is even more pronounced during maneuvers. The maximum flapping recorded for the radial-thrust configuration was ±4 degrees. The average maximums for the UH-1C and the previous elastomeric bearing tail rotors, during similar conditions, are approximately ±7 degrees. The reduction in flapping is believed to be the result of a blade motion coupling with blade pitch change. This coupling consists of blade beamwise motion about an effective hinge outboard of the pitch horn which introduces a blade pitch change that opposes the motion.

The pitch link oscillatory loads for this rotor are high when compared with standard data as shown by Figure 23. This increase in loads is attributed to mast whirl, which is caused by rotor out-ot-balance on a nonisotropic mount. Also, the blade pitch change loads feed through the elastomeric bearing blade grip (because of low friction) to the pitch link more than for the UH-1 blade grip. With the standard UH-1 grip bearings, the blade centrifugal force is supported by ball bearings, which are also rotated during blade pitch change. The friction in the ball bearings dampens the blade feedback loads. However, bearing race brinelling is believed to result from these load and motion conditions. The mast whirl, discussed below, also produces high mast oscillatory bending loads. However, the mast bending loads did not present a problem, since the mast design was predicated on other requirements more severe than the loads recorded.

A major part of the increase in mast and pitch link loads is attributed to an increase in the tail rotor mast natural bending frequency to a value near two-per-rev. An out-of-balance force will cause the rotor mast to follow an elliptical path. Corresponding to the mast whirl, the mast experiences steady

and two-per-rev parallel and perpendicular bending moments. Ordinarily, these moments are small; however, the closer the mast bending mode is to the two-per-rev forcing function, the greater the mast deflections and loads. The elliptical motion of the mast is accompanied by a two-per-rev inplane hub shear in the rotating system.

The effective mass of the standard UH-1 rotor is larger than the effective mass of the elastomeric rotor due to the virtual hinge of the latter. This causes an increase in pylon frequency. A static shake test of the UH-1 rotor revealed that the first mast bending mode was at 48 cps or 1.79-per-rev of the tail rotor. The first mode for the elastomeric bearing rotor is believed to be near two-per-rev.

A second effect of the apparent inplane hinge is increased inplane deflections. These inplane deflections are acted on by steady vertical shear loads which produce blade pitching moments, in this particular case at two-per-rev. These pitching moments are reacted at the pitch links; consequently, higher two-per-rev pitch link loads are experienced. The mid-span inertia weight (see Figure 17) shifts the blade weight chordwise (center of gravity) toward the blade trailing edge, which magnifies this condition. Therefore, the pitch link loads will be reduced somewhat, if the mid-span weight is installed at the blade quarter chord.

The elimination of the outboard stop in the grip and the provisions for the chevron thrust bearing configuration would reduce the effective mass of the hub and move the pylon natural frequency away from two-per-rev, thus reducing the pylon whirl magnitude. This reduction, which requires redesign, should result in lower shaft bending loads as well as a reduction in pitch link loads.

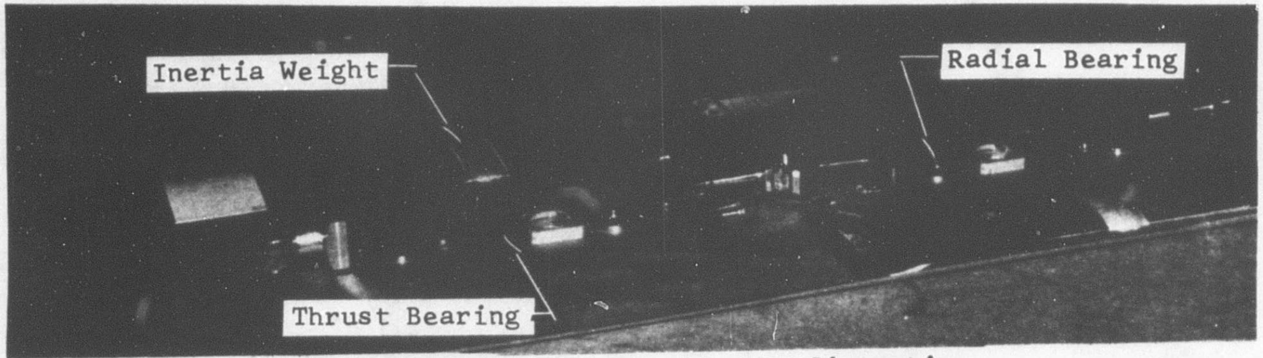
SECONDARY CONFIGURATION TEST RESULTS

The chevron bearing rotor configuration was assembled as shown in Figure 2a using standard UH-1 blades (no mid-span weights). Flight tests of this configuration were conducted as a part of Bell Helicopter Company's Independent Research and Development program.

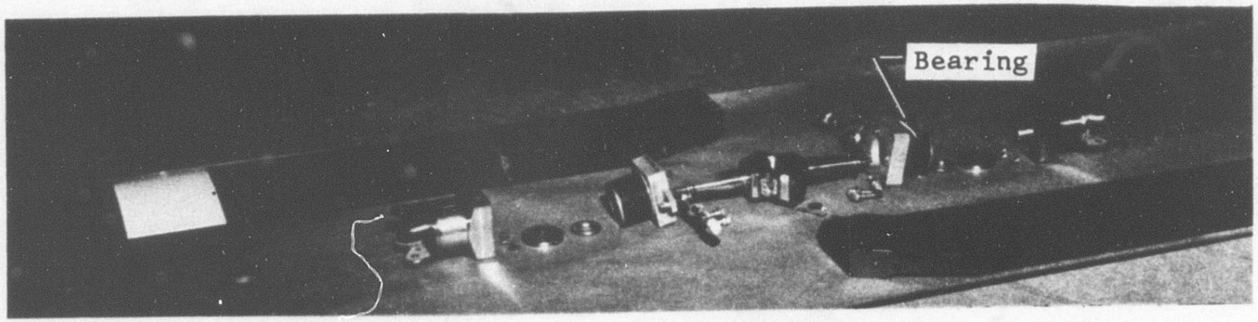
Rotor rotational frequencies were investigated as reported in the dynamics section of this report. Ground runs were completed, and the rotor appeared to be satisfactory for the inflight tests. Flight tests were conducted, and the rotor loads were monitored during hover, turns, acceleration, deceleration, and forward speed to 50 knots. Rotor "buzzing" was experienced during turns, acceleration, and deceleration flights as well as level forward flights. The beam and chord oscillatory loads recorded during the buzzing phenomenon were approximately twice comparable data for the radial-thrust bearing rotor.

The rotor buzzing is believed to be caused by blade pitch horn translation during pitch change, causing the blades to go out-of-track. The blade cyclic three-per-rev and collective yoke four-per-rev modes for this rotor are very near the forcing frequencies producing high oscillatory loads.

Since major changes were believed to be necessary to correct the oscillatory load problem, no further tests were conducted on this configuration.



a. Radial Plus Thrust Bearing Configuration



b. Chevron Bearing Configuration

Figure 17. Elastomeric Bearing Tail Rotors Before Assembly.

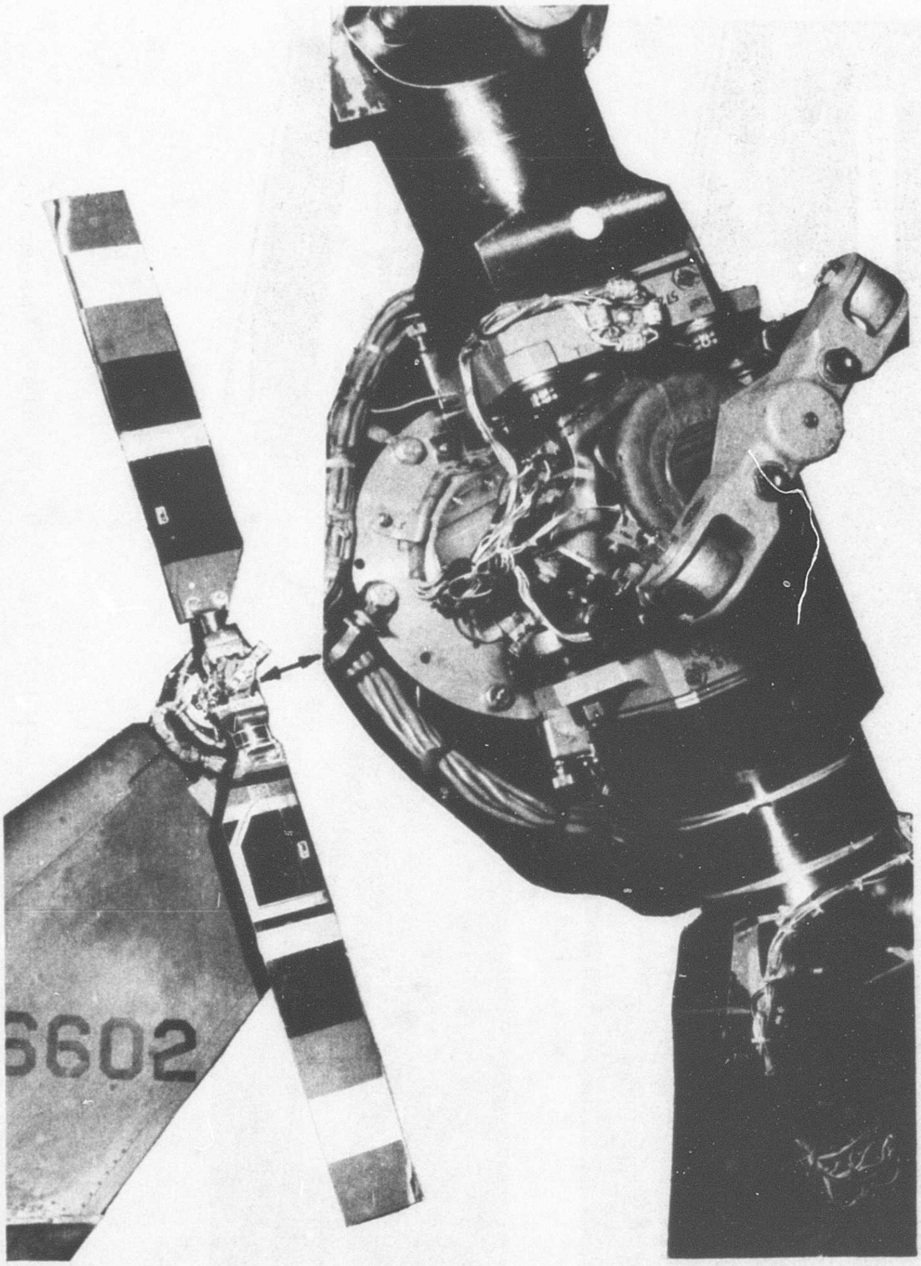


Figure 18. Elastomeric Bearing Tail Rotor Installed on the Test Helicopter.

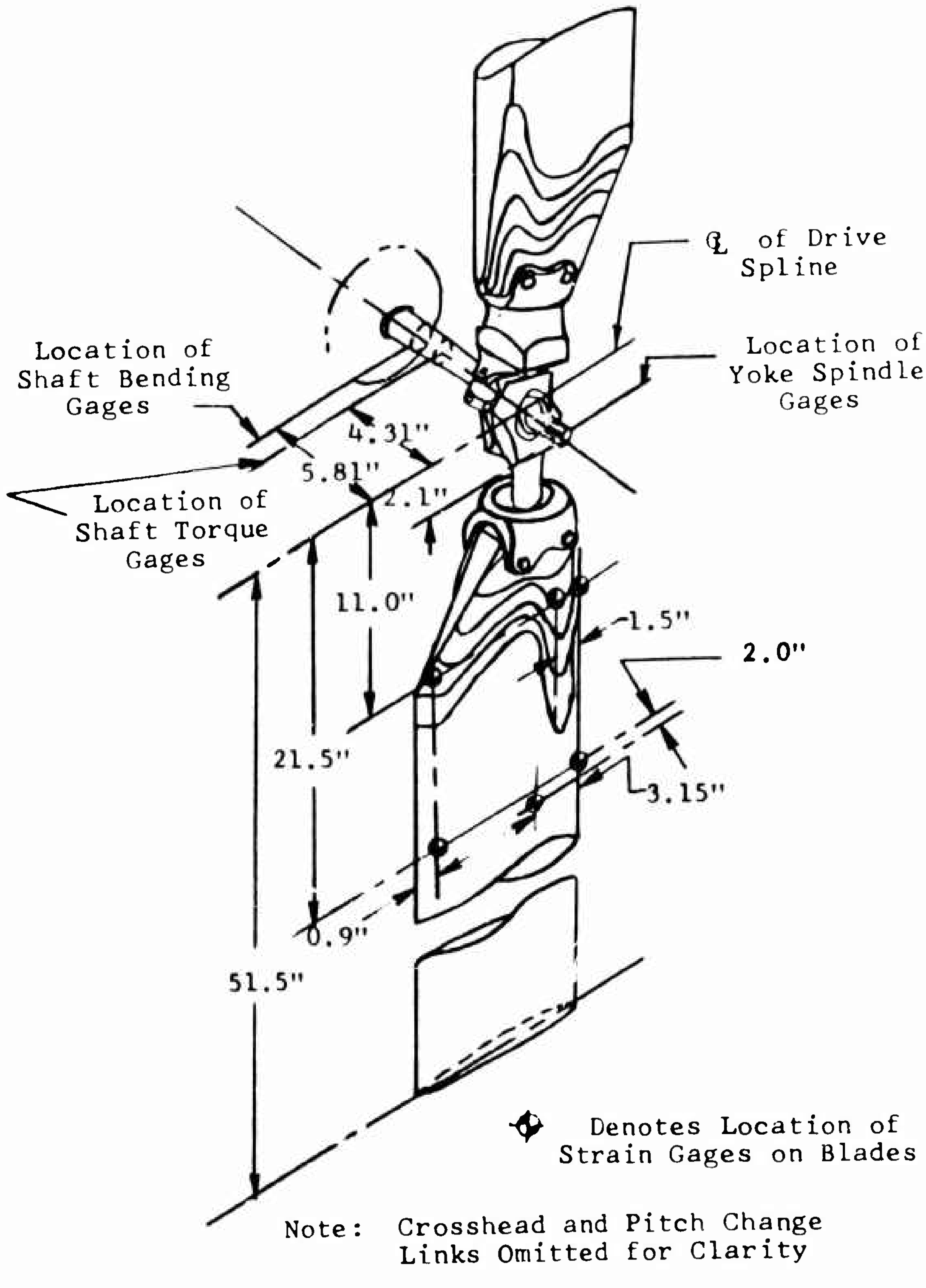


Figure 19. Tail Rotor Instrumentation Locations.

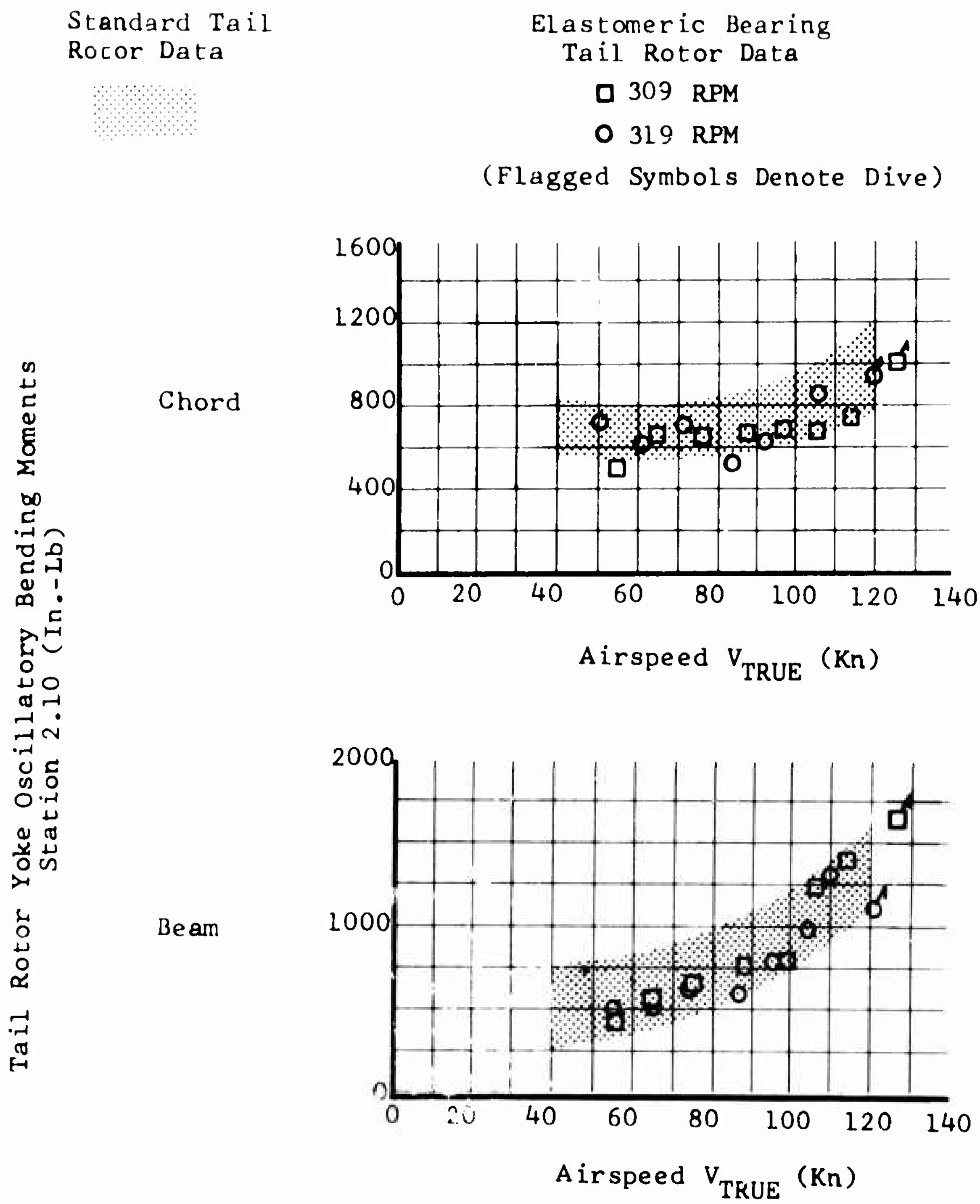


Figure 20. Tail Rotor Yoke Beam and Chord Oscillatory Bending Moments (Sta. 2.10) Versus Airspeed.

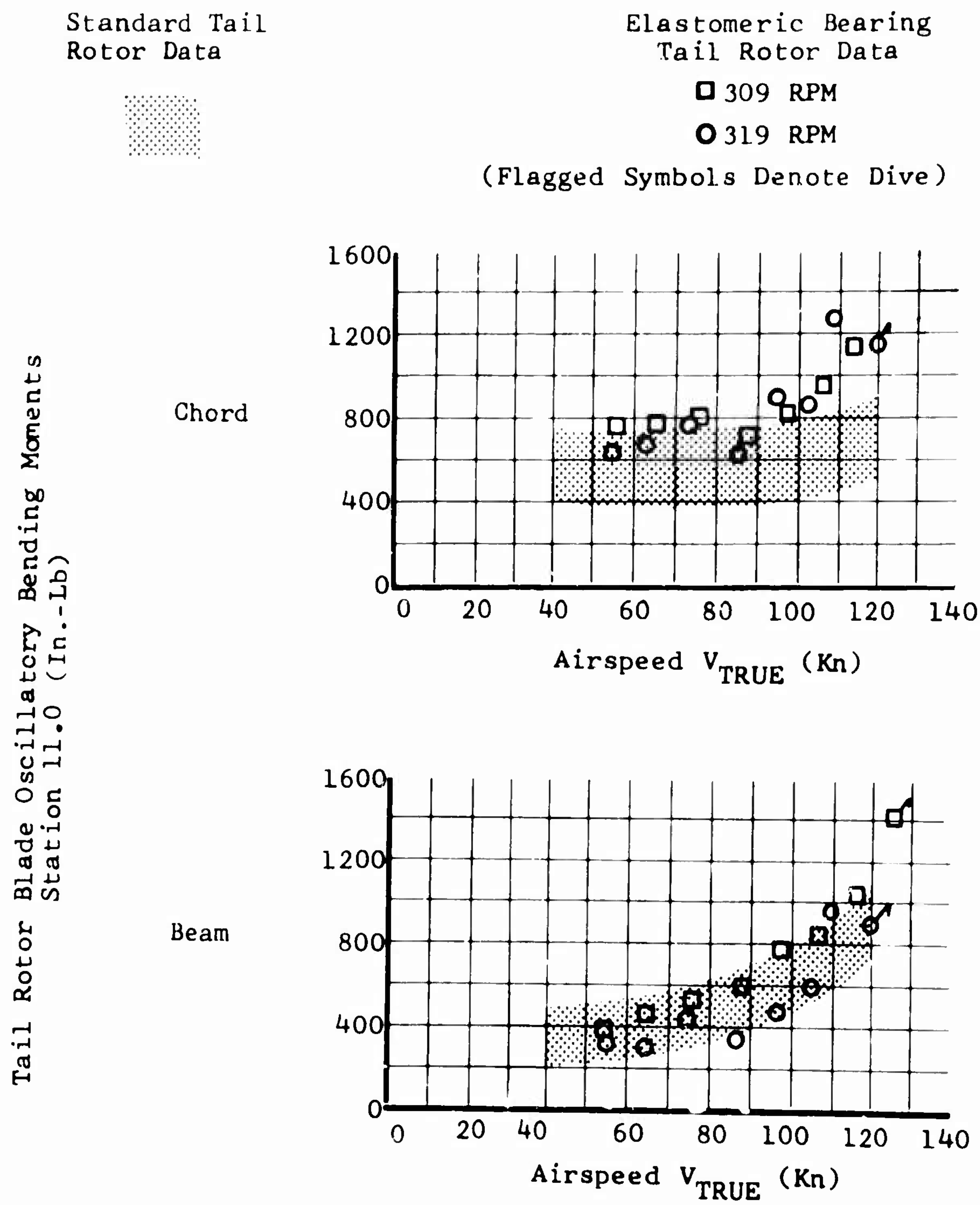
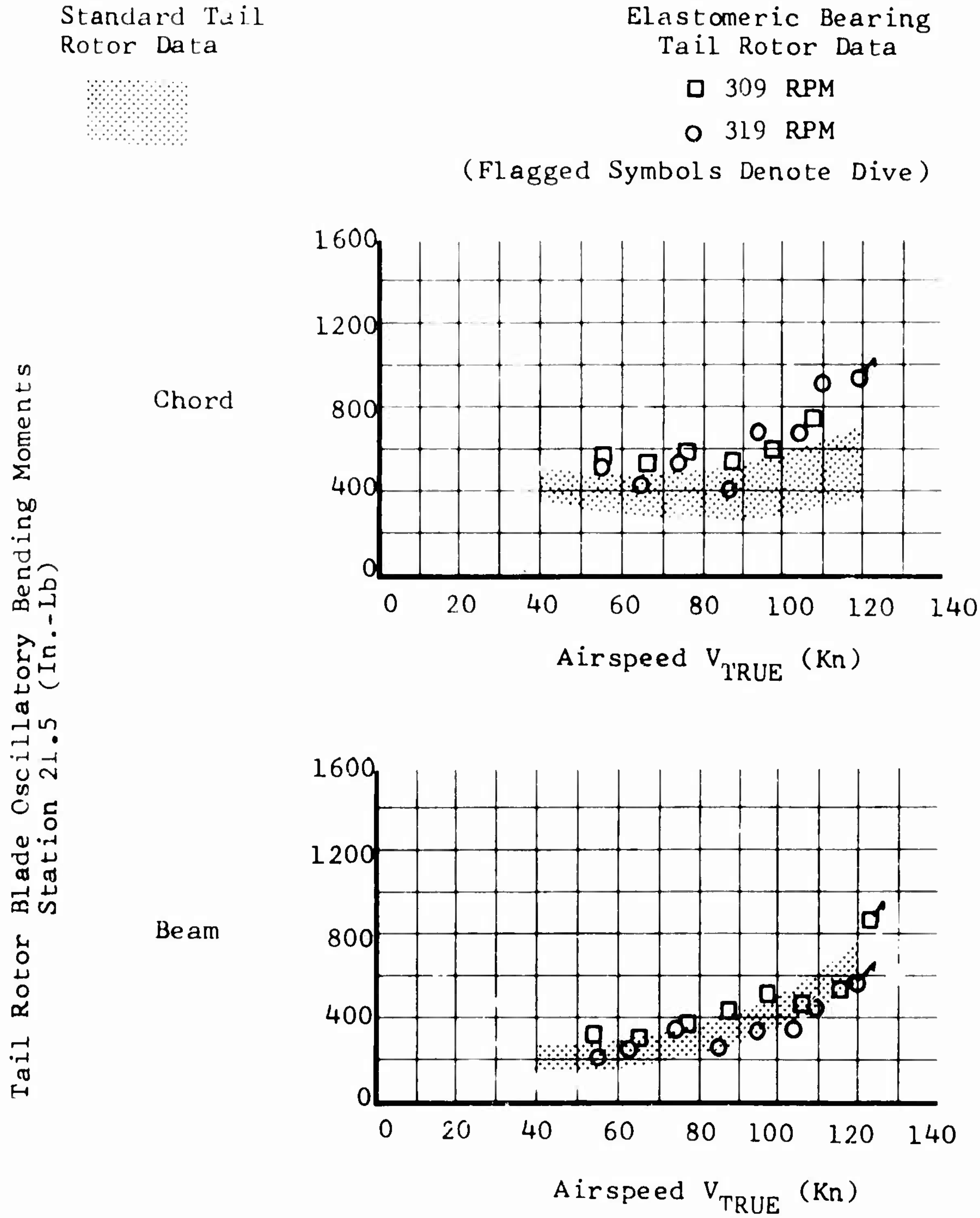


Figure 21. Tail Rotor Blade Beam and Chord Oscillatory Bending Moments (Sta. 11.0) Versus Airspeed.



(The Elastomeric Bearing Tail Rotor Beam Gage was located at Station 23.5)

Figure 22. Tail Rotor Blade Beam and Chord Oscillatory Bending Moments (Sta. 21.5) Versus Airspeed

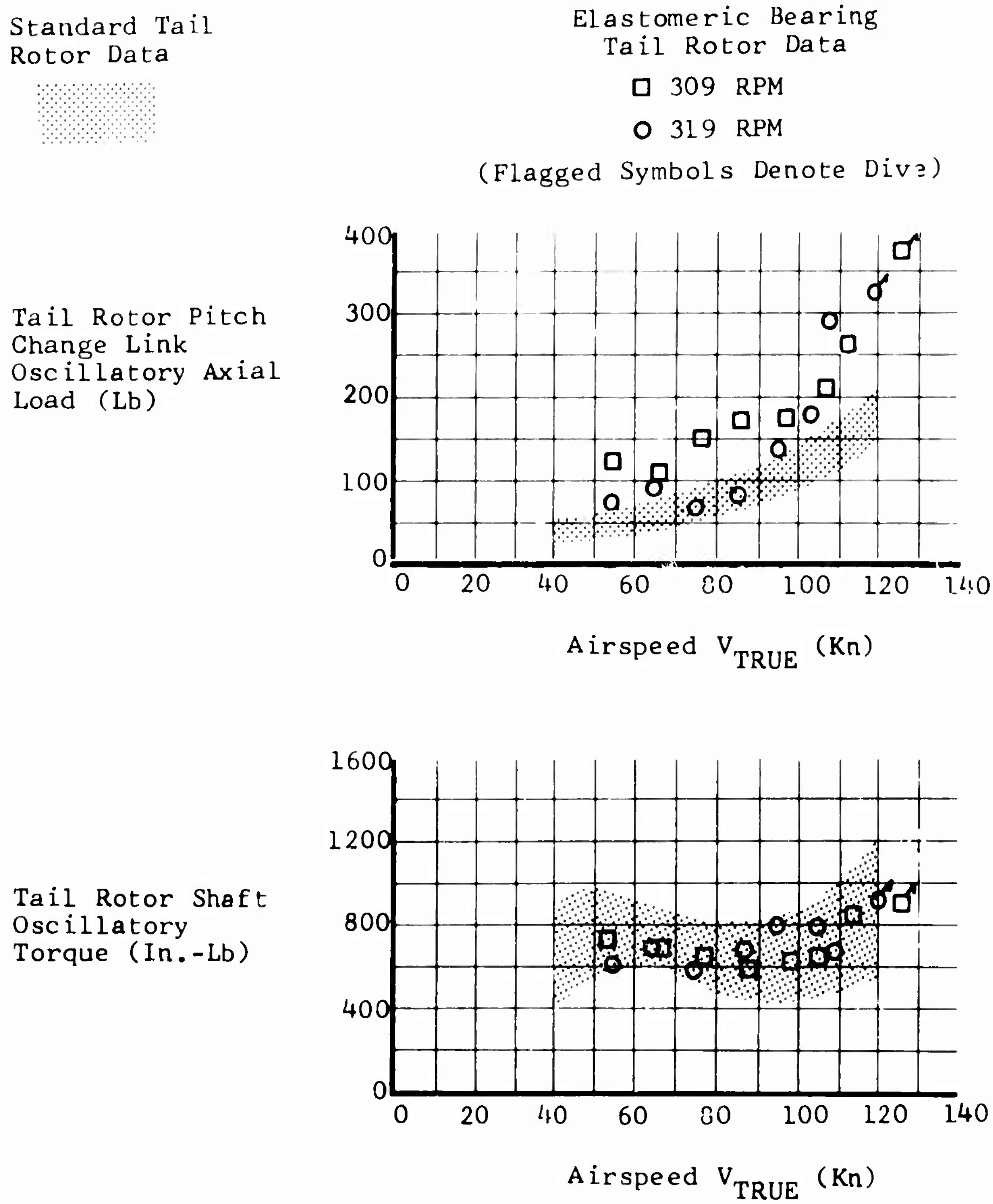


Figure 23. Tail Rotor Shaft Oscillatory Torque and Pitch Change Link Oscillatory Load Versus Airspeed.

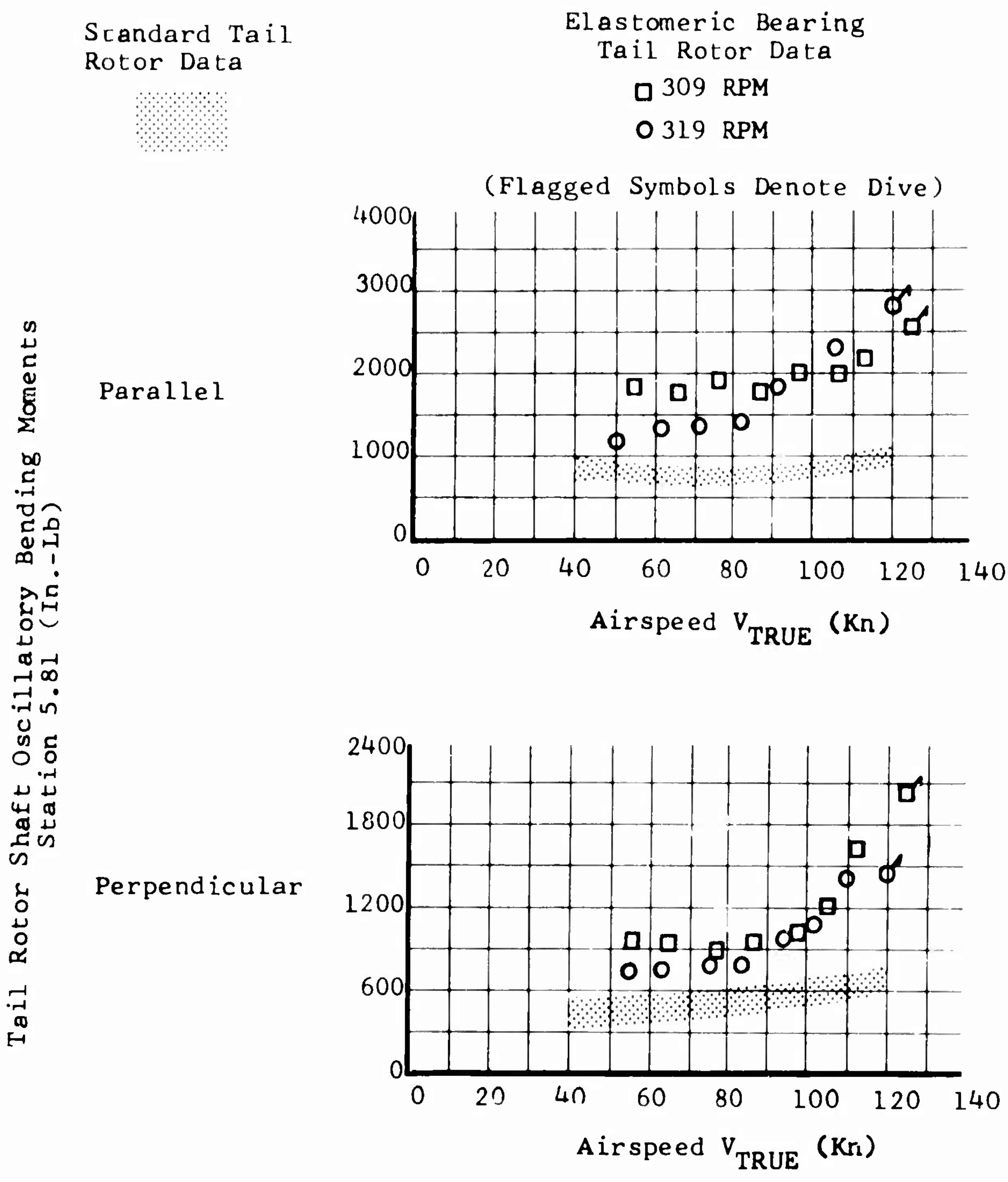


Figure 24. Tail Rotor Shaft Parallel and Perpendicular Bending Moments (Sta. 5.81) Versus Airspeed.

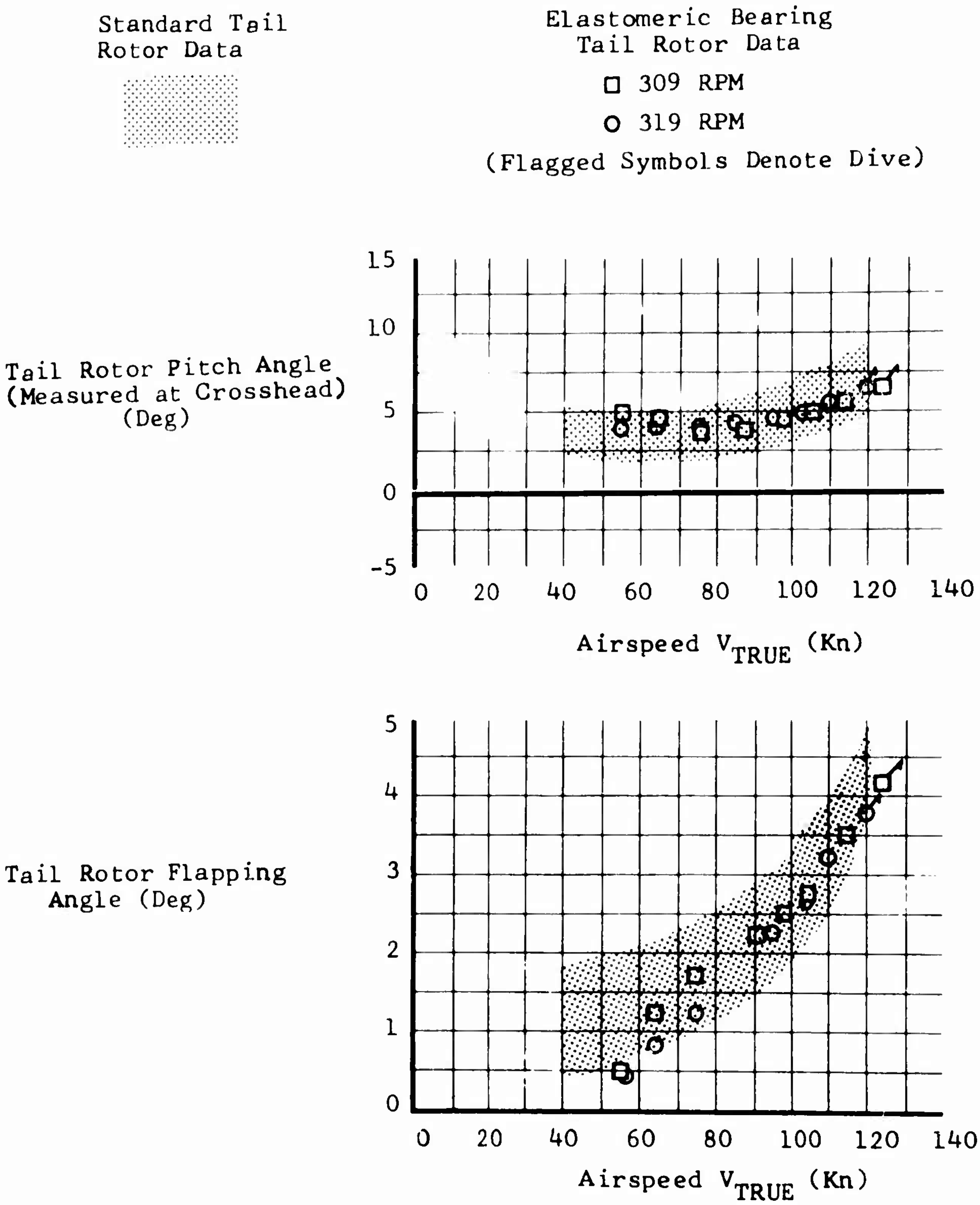


Figure 25. Tail Rotor Blade Pitch Angle and Rotor Oscillatory Flapping Angle Versus Airspeed.

CONCLUSIONS

The feasibility of using only elastomeric bearings in a helicopter tail rotor application has been proved. Of the two rotor configurations tested, the radial-thrust grip bearing arrangement was the most promising. With this configuration, the remaining questions and possible problems are related to hardware details and service use.

The expected advantages of an elastomeric bearing rotor consist of a reduction in maintenance (including simpler inspection and field replacement and the elimination of lubrication requirements) as well as a reduction in flapping and oscillatory loads. Also, the elastomeric bearing provides the designer with an additional tool to adjust the rotor natural frequencies.

Limited endurance tests indicated that elastomeric bearings will have a longer service life than conventional bearings; however, service tests are necessary to determine the magnitude of this advantage. Other unanswered technical questions and possible problems are the adverse effects of rotor balance and track, environmental effects on the elastomer and rotor weight when compared with conventional bearing rotors.

The above questions will remain open, for all elastomeric bearing applications, until a rotor system is fully developed and tested in service. This program has demonstrated that the tail rotor application is ideal for an elastomeric bearing rotor development and service test program.

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APPENDIX
TABULATED FLIGHT TEST RESULTS

TABLE I. INSTRUMENTATION

The following channels of tail rotor instrumentation were recorded on an oscillograph during the ground run and flight test program. Figure 19 presents the locations of the instrumentation strain gage bridges on the tail rotor shaft, yoke, and blade. For reference, the instrumented blade is identified as the "red" blade.

CHANNEL	SIGN CONVENTION FOR POSITIVE VALUES	UNITS
Yoke Spindle Sta. 2.1, Chord Bending	Blade leading edge in tension	in.-lb
Yoke Spindle Sta. 2.1, Beam Bending	Blade bends toward tail boom	in.-lb
Blade Sta. 11.0, Chord Bending	Blade leading edge in tension	in.-lb
Blade Sta. 11.0, Beam Bending	Blade bends toward tail boom	in.-lb
Blade Sta. 21.5 Chord Bending	Blade leading edge in tension	in.-lb
Blade Sta. 23.5 Beam Bending	Blade bends toward tail boom	in.-lb
Pitch Link (Red) Axial Load	Tension	lb
Shaft Torque	Blade leading edge in tension	in.-lb
Shaft Perpendicular Bending	Side of shaft toward leading edge of instrumented blade in tension	in.-lb
Shaft Parallel Bending	Shaft bends toward instrumented blade	in.-lb
Flapping Position	Instrumented blade toward tail boom	deg

TABLE I - Continued

CHANNEL	SIGN CONVENTION FOR POSITIVE VALUES	UNITS
Blade Angle*	Leading edge toward tail boom	deg

*NOTE: The blade angle was measured at the pitch change crosshead and will not monitor the effect of δ_3 (pitch change with flapping).

TABLE II. LEVEL FLIGHT LOADS AND DISPLACEMENT DATA

Model UH-1C 1584 Ship AF66-602	G.W. = 7500 Lbs H_D = 3000 Ft (approx.)	Flt. 136A 25 May 1968

CTR NO.	TEST CONDITIONS	MAIN ROTOR RPM	V_{TRUE} (KNOTS)
765	Level Flight	319	53.6
766	Level Flight		62.8
767	Level Flight		74.2
768	Level Flight		85.5
769	Level Flight		94.8
770	Level Flight		103.0
771	Level Flight		109.7
772	Dive	319	120.5
773	Level Flight	309	55.2
774	Level Flight		65.3
775	Level Flight		76.4
776	Level Flight		88.1
777	Level Flight		97.7
778	Level Flight		106.2
779	Level Flight		113.6
780	Level Flight	309	124.2

CTR NO.	T/R YOKE CHORD AT 2.1		T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0	
	MEAN	OSC	MEAN	OSC	MEAN	OSC
765	288	704	-1885	582	-528	625
766	224	608	-1875	611	-673	673
767	144	688	-1835	671	-673	769
768	192	514	-1737	690	-817	625
769	736	640	-1510	799	-480	865
770	736	709	-1371	977	-384	865
771	544	833	-740	1332	-192	1250
772	720	913	-750	1105	-96	1154
773	320	481	-1708	503	-481	769
774	384	641	-1648	622	-673	769
775	433	657	-1530	642	-529	817
776	224	673	-1471	780	-433	721
777	625	657	-1421	809	-48	817
778	449	673	-997	1234	0	962
779	849	753	-711	1402	0	1154
780	609	1025	10	1668	-	-

TABLE II - Continued

CTR NO.	T/R BLADE BEAM AT 11.0		T/R BLADE CHORD AT 21.5		T/R BLADE BEAM AT 23.5	
	MEAN	OSC	MEAN	OSC	MEAN	OSC
765	0	329	-1979	517	-86	215
766	-70	306	-2070	426	-140	269
767	-70	447	-2070	548	-161	356
768	23	353	-2101	395	-97	269
769	70	494	-1705	669	-10	334
770	47	612	-1522	699	-53	356
771	306	965	-1674	883	75	463
772	306	918	-1583	913	140	550
773	-71	400	-1888	548	22	302
774	-94	470	-2010	548	-86	281
775	-71	541	-1857	579	-11	356
776	-24	589	-1705	548	86	432
777	-24	777	-1462	609	140	507
778	188	848	-1462	731	97	464
779	471	1036	-	-	108	518
780	306	1436	-	-	162	874

CTR NO.	T/R PITCH LINK RED		T/R SHAFT TORQUE		T/R SHAFT PERP. BEND	
	MEAN	OSC	MEAN	OSC	MEAN	OSC
765	38	76	1122	385	746	811
766	19	96	1106	402	746	811
767	9	67	888	351	713	843
768	-38	76	938	402	811	811
769	9	144	1089	486	778	973
770	28	182	1089	486	1070	1070
771	96	288	1424	452	941	1395
772	96	326	1491	553	1070	1395
773	0	115	1106	469	584	974
774	10	106	972	402	487	941
775	10	144	888	419	714	909
776	19	173	855	352	746	941
777	19	173	972	369	1038	1038
778	38	211	1089	385	1071	1266
779	67	259	1156	520	876	1525
780	96	384	1408	536	1006	2044

TABLE II - Continued						
CTR NO.	T/R SHAFT PARA. BEND		T/R HUB FLAPPING		T/R BLADE ANGLE	
	MEAN	OSC	MEAN	OSC	MEAN	OSC
765	60	1089	-	0.6	3.8	-
766	-121	1271	-	0.9	3.9	-
767	-60	1271	-	1.3	3.8	-
768	-363	1392	-	1.7	3.9	-
769	-484	1815	-	2.3	4.3	-
770	-544	1997	-	2.7	4.7	-
771	-635	2209	-	3.3	5.6	-
772	-635	2814	-	3.8	6.1	-
773	-212	1846	-	0.6	4.7	-
774	-182	1816	-	1.3	4.0	-
775	-333	1097	-	1.8	3.5	-
776	-272	1786	-	2.2	3.6	-
777	-303	1937	-	2.4	4.5	-
778	-182	1937	-	2.8	4.9	-
779	-605	2119	-	3.5	5.8	-
780	-878	2573	-	4.0	6.5	-

TABLE III. MANEUVER LOADS AND DISPLACEMENT DATA

Model UH-1C 1584 Ship AF66-602	G.W. = 7500 Lbs H_D = 2000 Ft	Flt. 136B 25 May 1968

CTR NO.	TEST CONDITION	MAIN ROTOR RPM	V_{TRUE} (KNOTS)
784	Hover	319	0.0
785	Hover	309	0.0
786	Hover-Lt Turn-Stop	319	0.0
787	Hover-Rt Turn-Stop	↑	0.0
788	Lt Sideward Flight		36.0
789	Rt Sideward Flight		36.0
790	Full Power Climb		52.5
791	Full Power Climb		69.5
792	Full Power Climb		85.5
793	Left Turn		103.0
794	Right Turn		103.0
795	Autorotation		59.7
796	Autorotation	↓	77.8
797	Autorotation	319	95.8

CTR NO.	T/R YOKE CHORD AT 2.1		T/R YOKE BEAM AT 2.1		T/R BLADE CHORD AT 11.0	
	MEAN	OSC	MEAN	OSC	MEAN	OSC
784	534	534	-562	424	-450	550
785	660	597	-138	690	-600	801
786	1006	786	365	720	-400	1101
787	943	911	848	829	-100	901
788	47	958	-473	908	-550	951
789	1383	1540	888	1046	-350	1652
790	361	707	-720	602	-500	801
791	314	597	-720	878	-750	750
792	393	864	-700	858	-1051	1351
793	282	817	-1401	927	-1051	951
794	235	854	-1589	1016	-650	951
795	125	628	-3513	769	-1101	700
796	62	691	-3247	444	-600	700
797	47	707	-3109	424	-300	801

TABLE III - Continued

CTR NO.	T/R BLADE BEAM AT 11.0 MEAN	T/R BLADE BEAM AT 11.0 OSC	T/R BLADE CHORD AT 21.5 MEAN	T/R BLADE CHORD AT 21.5 OSC	T/R BLADE BEAM AT 23.5 MEAN	T/R BLADE BEAM AT 23.5 OSC
784	690	506	-2327	537	280	431
785	598	690	-2148	596	280	625
786	897	621	-2148	716	388	517
787	1174	897	-1939	686	647	582
788	322	966	-2327	775	43	820
789	966	920	-2148	1193	366	690
790	621	437	-2238	567	118	356
791	621	621	-2417	567	151	410
792	529	621	-2387	716	86	345
793	276	690	-2357	686	53	507
794	69	667	-2298	805	-21	496
795	-736	552	-2536	567	-291	334
796	-575	391	-2089	537	-302	258
797	-575	345	-1999	567	-258	237

CTR NO.	T/R PITCH LINK RED BEND	T/R PITCH LINK RED OSC	T/R SHAFT TORQUE MEAN	T/R SHAFT TORQUE OSC	T/R SHAFT PERP BEND MEAN	T/R SHAFT PERP BEND OSC
784	129	111	2324	298	506	760
785	139	157	2523	365	-63	950
786	204	185	2872	647	253	887
787	259	204	1843	448	633	1140
788	-129	166	547	547	475	1108
789	231	306	2324	730	538	1615
790	102	83	1809	481	443	887
791	92	92	1726	564	411	792
792	120	139	1328	697	696	1203
793	46	213	1278	614	570	1267
794	55	204	946	547	253	1267
795	-18	222	763	498	316	1203
796	-0	148	597	464	190	887
797	-9	102	714	415	316	1013

TABLE III - Continued

CTR NO.	T/R SHAFT PARA. BEND		T/R HUB FLAPPING		T/R BLADE ANGLE	
	MEAN	OSC	MEAN	OSC	MEAN	OSC
784	177	1301	-	0.6	10.1	-
785	-118	1715	-	0.7	11.2	-
786	236	1360	-	1.2	13.8	-
787	-117	1479	-	1.3	15.4	-
788	-266	1567	-	2.9	9.2	-
789	-236	1419	-	1.8	15.9	-
790	207	1331	-	1.3	8.0	-
791	-0	1656	-	1.9	7.0	-
792	88	2277	-	2.8	6.6	-
793	-236	2070	-	2.8	5.0	-
794	-118	1952	-	2.6	2.5	-
795	-354	1893	-	0.6	-1.7	-
796	-266	1390	-	0.7	-0.7	-
797	118	1419	-	0.7	-0.7	-

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13. ABSTRACT

This report presents the results of a research program to extend the state of the art of elastomeric bearings and to investigate further their application on a UH-1 tail rotor. In a previous program, Teflon journal bearings were used in conjunction with an elastomeric thrust bearing for carrying the edgewise and flapwise bending moments and for stiffness control. For the subject investigation, the Teflon journal bearings were eliminated. One radial and two thrust bearings were designed and fabricated for the blade grip application. One thrust bearing employed chevron shaped separators, and the other used 45-degree dished sheets between the layers of elastomer. Bench tests were conducted, and two blade grip bearing arrangements were whirl tested. One arrangement used a single chevron thrust bearing, and the second used the 45-degree dished thrust bearing in conjunction with the radial bearing. The tests showed acceptable frequency placement for the radial-thrust bearing arrangement. The chevron arrangement frequency placement was unsatisfactory. Flight tests of the radial-thrust bearing arrangement showed acceptable blade and hub structural loads, which were in most cases comparable to those of the UH-1. The rotor mast and control oscillatory loads were found to be higher than those of the UH-1. As in the previous program, the radial (flapping hinge) elastomeric bearing proved to be satisfactory. It is concluded that the feasibility of the elastomeric bearing tail rotor has been proved.

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Elastomer						
Helicopter						
Tail Rotor						
UH-1						

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